AERONAUTICAL ENGINEERING SERIES
GROUND ENGINEERS

MECHANICAL TESTING OF METALLIC MATERIALS

WITH SPECIAL REFERENCE TO PROOF STRESS

BY

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SECOND EDITION

LONDON
SIR ISAAC PITMAN & SONS, LTD.
1945

Second Edition, 1944
Reprinted, 1945

SIR ISAAC PITMAN & SONS, LTD.
PITMAN HOUSE, PARKER STREET, KINGSWAY, LONDON, W.C.2
THE PITMAN PRESS, BATH
PITMAN HOUSE, LITTLE COLLINS STREET, MELBOURNE
UNITEERS BUILDING, RIVER VALLEY ROAD, SINGAPORE
27 BECKETTS BUILDINGS, PRESIDENT STREET, JOHANNESBURG

ASSOCIATED COMPANIES
PITMAN PUBLISHING CORPORATION
2 WEST 45TH STREET, NEW YORK
205 WEST MONROE STREET, CHICAGO
SIR ISAAC PITMAN & SONS (CANADA), LTD.
(INCORPORATING THE COMMERCIAL TEXT BOOK COMPANY
PITMAN HOUSE, 381-383 CHURCH STREET, TORONTO

BOOK
PRODUCTION
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STANDARD

THE PAPER AND BINDING OF
THIS BOOK CONFORM TO THE
AUTHORIZED ECONOMY STANDARDS

MADE IN GREAT BRITAIN AT THE PITMAN PRESS BATH
D5—(A.70)

PREFACE TO SECOND EDITION

In this new edition opportunity has been taken to rewrite or to revise much of the text. Chapter III has been rearranged to incorporate details and illustrations of the latest machines available.

The subject of proof stress remains one of the principal features, and the text has been amplified to include further details which will assist in the understanding of this extremely important test. The four-point method is treated in detail as no method of proof stress determination can compare with it when used for routine testing.

The essentially practical nature of the book is fully retained as it is this feature which has proved of the greatest value to readers.

R. A. B.

PREFACE

This book is based upon lectures and practical demonstrations given by the author at the Mechanical Testing Laboratory of the College of Aeronautical Engineering.

Many other books dealing with mechanical testing being available, it might be wondered what useful purpose could be served by adding yet another to the list. If another book had been available dealing with British Standard Specification tests in a practical manner, suitable for those wishing to learn fundamentals, this work would not have been attempted. It has not yet been possible to obtain, in any one publication, precise information about, and an appreciation of, the various factors involved in the carrying out of such routine tests as are called for in the British Standard Specifications. This is particularly the case when considering proof stress.

In this book, therefore, the subject of proof stress has been dealt with in detail, as there is very little information available on so important a test. It is hoped that this feature will be welcomed by test-house personnel.

The true function of the elastic modulus in its relation to practical testing has been clearly defined, as such use forms the very foundation of all intelligent testing, and leads to a better understanding of the factors involved in proof stress calculations.

It is anticipated that this manner of presentation will be of benefit to those who have no previous knowledge of the subject, as well as those more familiar with testing generally.

Descriptions of testing machines have been included in order that those not conversant with their use should be able to understand the method of operation, and also for a reference to be made when dealing with various features associated with individual machines.

The chapter dealing with X-ray examination and crack detection has been incorporated in order that aircraft engineers, and others who require an elementary knowledge of these non-destructive types of tests, might have this information included in the one book.

Finally, thanks are due to those firms who so readily supplied diagrams and information about their products, without which this book would not have been complete, and to Mr. S. Rees-Jones, to whom the author is indebted for the explanation of the principles of radiological testing.

R. A. B.

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MECHANICAL TESTING OF METALLIC MATERIALS

CHAPTER I

THE TENSILE TEST

THE test figures usually called for in British Standard Specifications for Aircraft Materials consist of the following—

(*a*) Maximum stress.
(*b*) Proof stress.
(*c*) Elongation, per cent.
(*d*) Reduction of area, per cent (less frequently).
(*e*) Hardness number.
(*f*) Impact value.

The first four values are obtained from a tensile test, and this test will be considered first, as it will also provide additional information if desired.

General Considerations of the Tensile Test. If a bar of hot-rolled mild steel be subjected to a steadily increasing load in the direction of its longitudinal axis and the increase in length due to such load is measured, it will be found that, up to a certain load, the extension produced will be directly proportional to the load, i.e. if the load is doubled, the extension will also be doubled. Continuing the test, a load will be reached at which the extension increases more rapidly, until suddenly a very rapid extension takes place. If still more load is applied, the extension will continue to increase, until finally the bar will not support additional load. After this point considerable extension will occur and will be accompanied by a "waisting" of the material, and fracture of the bar will result.

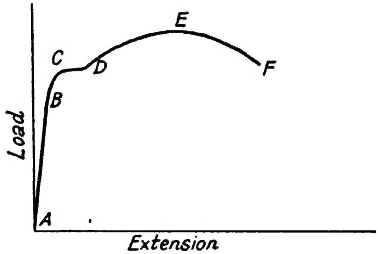


FIG. 1. LOAD/EXTENSION CURVE FOR MILD STEEL

By plotting the values of load and extension as shown in Fig. 1 the sequence of events can be clearly followed. The straight line from A to B represents that part of the test where the extension remains proportional to the load, B to C shows where the extensions increase more rapidly, and C is the point of sudden rapid extension, represented by C to D. The amount of extension up to point B is very small, and sensitive instruments are necessary to measure it.

After D, with further increase in load and extension, the point E of maximum load is reached. From E considerable local extension occurs

and "waisting" takes place, the cross-sectional area of the bar being greatly reduced. Due to this, a lesser load than at E is required to continue the elongation, and with decreasing cross-sectional area the load decreases until fracture occurs at F, E to F indicating the continued extension with decreasing load.

Particular names are given to these various points or parts of the curve, and are as follows: The straight line AB is termed the *elastic line* or *line of proportionality,* because up to the point B the bar would return to its original dimensions if the load were removed, and the material is said to behave elastically. If the material is loaded beyond B, on removal of the load a permanent extension would remain. The point B is known as the *limit of proportionality* or *elastic limit,* C as the *yield point,* E as the *maximum load,* F as the *breaking load.* The percentage elongation and reduction of area are obtained from measurements of the test bar after fracture.

Many metallic materials have no clearly defined yield point, and therefore the value of the proof stress is taken as a basis for stress calculations from the designer's point of view. The proof stress will be considered in detail later on.

For ordinary testing requirements to ascertain the values of maximum stress, yield stress (if any), percentage elongation, and reduction of area, it is not necessary to measure any extensions while the material is under load; but for more detailed information, such as is necessary for the determination of proof stress, such readings of extensions must be taken.

DETAILED CONSIDERATIONS OF THE TENSILE TEST

Stress and Strain. Although the load extension scale is often used for plotting parts of the complete curve as shown in Fig. 1, the values of stress and strain are also employed.

The term *stress* in the accepted meaning refers to "intensity of stress," which is the load per unit area, the unit area being 1 sq. in., therefore—

$$\text{Stress} = \text{load}/\text{area}$$

The term *strain* is defined as the extension per unit length. If, therefore, extensions are measured over a 2 in. length of the test bar, known as the gauge length, the strain will be the numerical value of the extension divided by 2. For example, an extension of 0·002 in. on a 2 in. gauge length is equivalent to a strain of 1/1000, i.e. an extension of 0·001 in. per in. of gauge length. The strain value is a mere number, as it is the result of dividing extension in inches by gauge length in inches.

HOOKE'S LAW AND THE ELASTIC MODULUS

This law states that within the limits of elasticity stress is proportional to strain, or in other words—

$$\text{Stress}/\text{strain} = \text{a constant}$$

As the strain is a mere number, it follows that the dimensions of the constant will be the same as for the stress, i.e. in tons or pounds per square inch. The constant in the equation is called the *elastic modulus* or

the *modulus of elasticity* for the particular material to which the stress and strain refer.

If part of the curve of Fig. 1 representing the elastic line be drawn on a stress/strain basis to a larger scale, as in Fig. 2, and it is found that the value of the strain AD for a stress AC 12·5 tons/sq. in. is 0·001, then by Hooke's Law—

$$\frac{12 \cdot 5}{0 \cdot 001} \text{ tons/sq. in.} = \text{the elastic modulus}$$

or

$$12{,}500/1 = 12{,}500 \text{ tons/sq. in.}$$

The elastic modulus can therefore be defined as that stress which would cause unit strain if the material were to remain elastic.

The value of the modulus can only be applied within the limits of elasticity, i.e. to the extent of the straight line of the graph. If in the figure DE is substituted for AC, the elastic modulus will be given by DE/AD, and this is then seen to be the gradient of the elastic line, so that the value of the constant will be unchanged if the stress $D'E'$ divided by the equivalent strain $A'D'$ is taken or any other such triangle on the line. Practically, therefore, the value of the elastic modulus for any given material can be ascertained from the elastic line of the plotted load extension, or stress/strain diagram. If the diagram is load-extension it is only necessary to take a similar triangle on the elastic line and to convert the load to stress, and the equivalent extension to strain, to obtain the modulus directly.

The Practical Importance of the Elastic Modulus. The value of the elastic modulus is practically constant for any given material. For the mild steel graph of Fig. 2 the figures given were average figures which might be obtained during such a test, and it was seen the modulus had a value of 12,500 tons/sq. in. That value is an average value for any normal steel, whether plain carbon or alloy, normalized or heat-treated. The value being constant, it follows that the slope of the elastic line will be constant, and this fact is a means of check on the results obtained from a test.

Different metallic materials have their own particular values for the elastic modulus, and knowledge of these enables the engineer to identify the material from a graph embracing the elastic line only. This feature is illustrated in Fig. 3. Another most important feature is the use of the elastic modulus to determine whether a test is being satisfactorily carried out, and in fact no intelligent testing of materials can be carried out without such use. If, for example, a load/elongation graph for some material is to be obtained, and the test bar, with the instrument to read the extensions attached, is under load in the testing machine and extensions against load are being recorded, how is the operator to know if some unforeseen circumstance, whether connected with the testing machine, the extensometer or the operator's own miscalculation is at work, rendering the extensometer readings entirely incorrect? Unless the application of Hooke's Law is known it will not usually be possible to tell.

This is the practical application. Knowing the material which has to be tested and the average value of its elastic modulus, *calculate* the

extensions which *should* be produced by the increments of load that will be used in the actual test.

For example, it is intended to load an alloy steel test bar of 0·25 sq. in. section, 2 in. gauge length, with 1 ton increments, and to note the corresponding extensions for the purpose of plotting a graph.

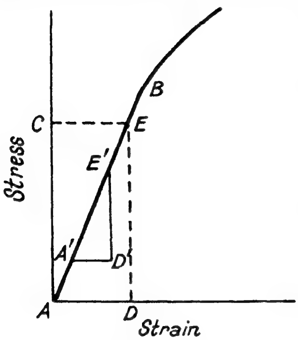


FIG. 2. THE ELASTIC LINE OF THE STRESS/STRAIN CURVE

Taking 12,500 tons/sq. in. as an average value for the elastic modulus, use the equation—

$$\text{Stress/strain} = \text{modulus}$$

The stress intensity due to 1 ton load on 0·25 sq. in. cross-sectional area will be—

$$1/0{\cdot}25 = 4 \text{ tons/sq. in.}$$

The strain will be the amount of extension divided by the gauge length of 2 in. Thus—

$$\frac{\text{Stress}}{\text{Strain}} = 12{,}500$$

$$\frac{4 \text{ tons/sq. in.}}{\dfrac{\text{Extension (in.)}}{2 \text{ in.}}} = 12{,}500 \text{ tons/sq. in.}$$

$$\begin{array}{r}\text{Extension on 2 in. gauge}\\ \text{length per 4 tons/sq. in.}\end{array} = \frac{4 \times 2}{12{,}500} = 0{\cdot}00064 \text{ in.}$$

Thus it is seen that load increments of 1 ton, i.e. stress increments of 4 tons/sq. in., should produce elastic extension increments of 0·00064 in. This will hold true until the elastic limit of the material is reached.

The extensometer used to measure these extensions for ordinary requirements reads to 1/10,000 in. or 1/20,000 in. Taking each division on the extensometer dial as representing an extension of 1/20,000 in. (0·00005 in.), the 0·00064 in. calculated above will represent 12·8, say 13 dial divisions. The measured extension should not vary from that calculated by more than about minus 1·5 dial divisions for each 4 tons/sq. in. increment during the elastic range. Knowing this, therefore, it is a simple matter to check the actual extension obtained when the material is under test, and if it does differ more than this amount from the calculated value, stop the test and investigate. By thus calculating the extension which should occur, and checking the extension which does occur against it, there is no possibility of false readings due to error in the machine or the extensometer.

As previously mentioned, the value 12,500 tons/sq. in. is an average value for steels, and due to fluctuations in this figure the amount of extension will vary slightly for the same loading on different steels. Naturally, only practical experience in testing materials enables the operator to expect a little higher or lower reading than that calculated on the basis of 12,500 tons/sq. in. The important point is, however, that

even by using this average value it will be impossible to be led astray, and the operator does not have to depend absolutely upon the indications of the extensometer.

An error due to the extensometer consistently over or under reading can still produce a perfect-looking diagram when results are plotted, but the slope of the graph will be incorrect and the data obtained from such a diagram will be of little value, and once the elastic limit has been passed it will be impossible to restart the test. In other words, a knowledge of the practical application of Hooke's Law enables a constant check to be kept on the conditions of the test.

With reference to the 0·00064 in. extension per 2 in. gauge length of steel due to a stress intensity of 4 tons/sq. in., this is equivalent to 0·00016 in. extension per ton per square inch per 2 in. gauge length. It can thus be stated that steel will extend *elastically* at the average rate of 0·00008 in. per ton per square inch per inch of gauge length. It can be applied as follows.

For example, a steel test bar of 0·25 sq. in. cross-section and 2 in. gauge length might have to be checked by loading to a stress intensity of 30 tons/sq. in., with the proviso that the extension recorded at that stress must not exceed 0·0068 in. if the material were to be passed.

On conducting the test, imagine that the actual extension given was 0·0030 in. It might well be thought that, as 0·0030 in. certainly is far from exceeding the 0·0068 in. allowed, the test and the material would both be highly satisfactory. A little consideration, however, will show the test to be far from satisfactory, and indicate the presence of some cause that would entirely upset the readings obtained.

If steel extends elastically, 0·00016 in. per ton per square inch per 2 in. gauge length, at 30 tons/sq. in. the amount of elastic extension will be 0·0048 in. on the 2 in. gauge length. The only variation which could occur would be of the order of a few 1/20,000 in., and if the elastic limit had been exceeded the extension would be considerably more.

By such simple calculation it is seen to be an impossibility for normal steel to extend only 0·003 in. for a stress intensity of 30 tons/sq. in. The reason for such a low reading in this particular instance could have been due to the extensometer needle sticking. If Fig. 3 is plotted to a large scale using graph paper marked in one-tenth, half, and one-inch squares, it will be a simple matter to read off average elastic extensions for any given stress.

By intelligent use of the factor 0·00008 in. per ton per square inch per inch of gauge length, the extension should have been checked as the load was progressing and the abnormally low readings noted earlier. In case it should occur to the reader to ask how the given extension of 0·0068 in. could ever have been obtained for a stress intensity of 30 tons/sq. in., it must be stated that for the particular material in question 30 tons/sq. in. was a stress slightly in excess of the elastic limit, when the extensions are in excess of those obtained within the elastic range.

Further Application of Hooke's Law. The foregoing application of the stress/strain relationship is not confined to steel. The law holds good for other materials, and the extensions which should be obtained when testing non-ferrous alloys are simply, as with the steel, functions of their

particular elastic moduli and can easily be referred to the value of the extension for steel.

If a material has an elastic modulus one-half the value of that for steel, then its extension for each 1 ton/sq. in. stress intensity will be twice as

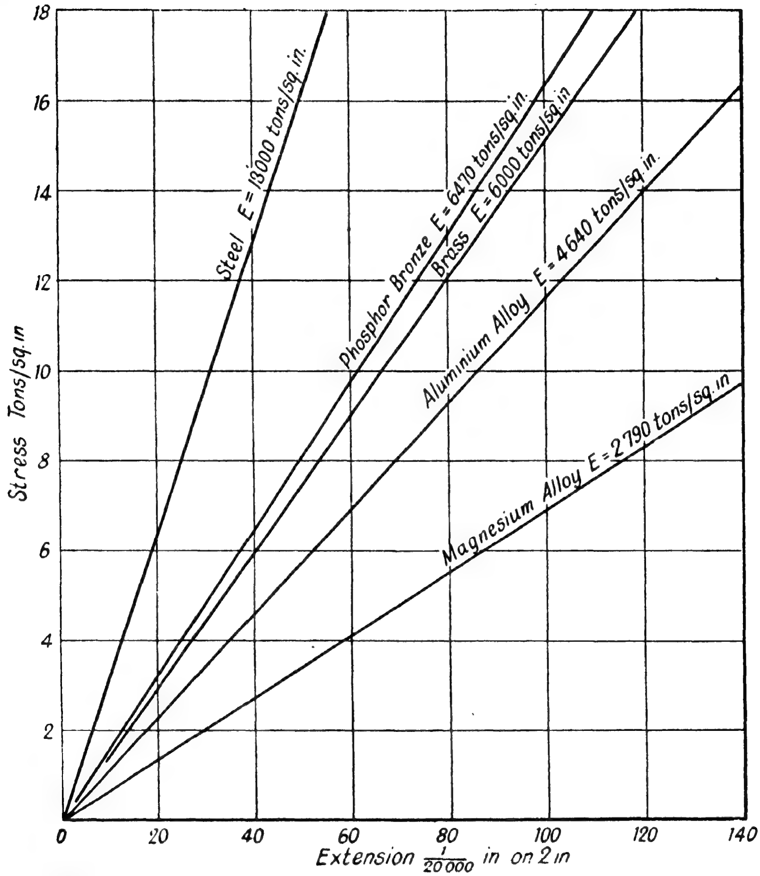


FIG. 3. STRESS EXTENSION IN RELATION TO ELASTIC MODULUS

much as the extension which would be given for steel on the same gauge length. That is, if a particular material has an elastic modulus of 6250 tons/sq. in., the extension for 1 ton/sq. in. stress intensity will be $2 \times 0{\cdot}00016$ in. $= 0{\cdot}00032$ in. on 2 in. Similarly, an elastic modulus equal to one-third that of steel will give an extension of $3 \times 0{\cdot}00016 = 0{\cdot}00048$ in. per ton per square inch per 2 in. These are, as before mentioned, average figures, but none the less dependable for checking the test of any material.

It will be seen from Table I that the extension varies inversely as the modulus.

TABLE I. MODULI OF ELASTICITY
AVERAGE VALUES

Material	Modulus	Extension/ton/sq. in. on 2 in. gauge length.
	(tons/sq. in.)	in.
Steel	13,000	0·00015
Aluminium alloys	4,640	0·00043
Magnesium alloys	2,790	0·00071
Phosphor-bronze	6,470	0·00031
Brass (wrought)	6,000	0·00033
,, (cast)	4,000	0·00049

MAXIMUM STRESS, PERCENTAGE ELONGATION, AND REDUCTION OF AREA

Maximum Stress. When a specification states that the maximum stress of a material is a certain figure, say for example 45 tons/sq. in., it is often thought by those not familiar with testing that 45 tons/sq. in. is the stress intensity necessary to break the test bar. With any ductile material such an impression is totally incorrect, as will be shown by the following considerations. The maximum stress is defined as the maximum load divided by the original cross-sectional area of the gauge length of the test bar.

Referring to Fig. 1 (page 1), the point E on the curve represents the maximum load reached during the test, and although the bar has elongated a certain amount, the cross-section over the gauge-length is approximately uniform and does not differ materially from the original value. Therefore this load divided by the original area gives the maximum stress figure required by the specification.

Now, it has been stated that after point E is passed, the bar greatly extends, "waists," and fractures, and Fig. 1 shows a decreasing load after E. Obviously if E represents the maximum load which the bar will withstand with its sensibly original cross-section, when that cross-section is being rapidly reduced through much elongation and waisting a lesser load will suffice to continue the extension of the bar. Therefore a diagram plotted automatically by the testing machine will show the falling-off in load as in Fig. 1 until at F the bar fractures.

Now, if the actual load on the bar at the time of fracture is divided by the *reduced* cross-sectional area at fracture, the stress intensity will be higher than the figure given for the maximum stress. This breaking-stress figure is of no practical importance as regards the tensile test because the test bar has no longer its original cross-sectional area, and if the percentage elongation and reduction of area were not required there would be no need to continue the test after the maximum load had been reached.

Confusion often arises due to the curve of Fig. 1 being drawn with stress and strain as the ordinates. When this is done the breaking stress

is apparently lower than the maximum stress, and the person not familiar with testing may be perplexed as to the significance of the maximum stress figure when fracture apparently occurs, according to the graph, at a lower stress intensity.

Diagrams showing the full curve on a stress/strain or stress/extension basis are actually load/extension curves plotted automatically by the testing machine. For convenience in reading results the loads are converted into stress intensities, the values of which are printed on the chart instead of the actual loads. These stress values, calculated from the original cross-sectional area of the test bar, will cease to be correct when the cross-sectional area has altered, as it has substantially, after the maximum load. Consequently, although after the maximum load the autographic apparatus of the testing machine plots the reducing values of the load, the corresponding stresses cannot be taken from the existing chart scale which is only essentially correct up to the point of maximum load. Fig. 4 illustrates a load/extension curve, and in addition the true corresponding stress curve. The actual stress in the bar is that given by the upper broken line portion of the graph, obtained by measurement of the test bar at loads in excess of the yield point. It will be seen that the maximum stress value as required by specifications is really a nominal figure, as even at the maximum load the cross-section of the bar has reduced appreciably.

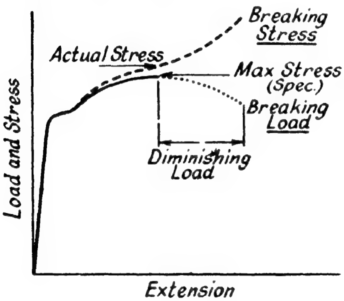


FIG. 4. LOAD AND STRESS EXTENSION CURVE

It should be noted that ultimate stress means the same as maximum stress, and the terms are used interchangeably to denote the maximum load divided by the original cross-sectional area.

There are cases when the breaking stress has essentially the same value as the maximum stress, and this occurs when testing cast material, etc., with little or no ductility. With these materials there is so little elongation that no decrease in cross-section occurs and the bar breaks at the maximum load. Therefore the breaking stress has essentially the same value as the maximum stress for these special cases.

Percentage Elongation. Percentage elongation of a test bar after fracture is expressed as—

$$\frac{\text{Increase in length of the gauge length}}{\text{Gauge length}} \times 100$$

The extension of a test bar under load has been mentioned earlier with reference to Fig. 1. Up to the maximum load this extension is sensibly uniform over the gauge length. For example, if gauge lengths of 1 in. and 2 in. be marked it will be found that if at the maximum load the 1 in. length has increased to 1·1 in., then the 2 in. length will have increased to 2·2 in., thereby giving the same percentage elongation. After the maximum load the ductile bar waists and then fractures. Now, the extension

of that length of the bar which includes the waist is considerably greater than that occurring elsewhere, so that the total extension after fracture over any particular gauge length will be made up of the general extension taking place up to the maximum load, and the large local extension at the waist.

If, for example, a 2 in. and 3 in. gauge length be symmetrically marked about the mid-portion of a test bar and the fracture occurs close to the mid-portion, the percentage extension will be greater on the basis of the 2 in. than that of the 3 in. gauge length. This is because most of the extension occurring at the neck will take place within the 2 in. length. It is therefore seen that the percentage elongation is dependent upon the particular gauge length used, and for standard testing this length must obviously be fixed. Also, supposing that two test bars of the same material but of different cross-sectional area are to be tested, if the material is the same in each case it is to be expected that the test results will compare in respect of the percentage elongation.

The question then arises as to what gauge lengths must be used to achieve this result. Clearly the same gauge length cannot be used on bars of different cross-sectional area, because the larger bar has a greater volume of material which can extend. In order, then, that tests on round bars may be truly comparable, the gauge length for British Standard Testing is based upon the fact that on cylindrical test pieces, if the ratio of the gauge length to the diameter or cross-sectional area of the bar is kept constant, then the percentage elongation will also be constant. This ratio is—

$$\text{Gauge length in.} = 4 \times \sqrt{(\text{cross-sectional area sq. in.})}$$
$$\text{or} \quad 3{\cdot}54 \times \text{diameter in.}$$

Knowing, then, the diameter of the test bar, it is an easy matter to calculate the appropriate gauge length to be used.

For the full value of the percentage elongation to be obtained the fracture should occur near the mid-point of the gauge length, in order that the local extension taking place should all be within the gauge points. It is for this reason that specifications state that should a tensile test piece break outside the middle half of the gauge length, another test may be made. This is in order that specimen shall not fail on account of too small a percentage elongation brought about by fracture taking place close to one gauge point. The middle half of the gauge length is marked as shown in Fig. 5.

Percentage Reduction of Area. This is expressed as—

$$\frac{\text{Original cross-sectional area} - \text{final cross-sectional area}}{\text{Original cross-sectional area}} \times 100$$

or

$$\frac{\text{Reduction in area}}{\text{Original area}} \times 100$$

The final cross-sectional area is calculated from measurement of the least diameter of the test bar after fracture. As the original cross-sectional area

of the test bar is invariably known, it is not necessary to solve the equation as it stands.

Taking, for example, a cross-sectional area of one-quarter of a square inch, which is the normal standard for aircraft material, the equation can be reduced to a more simple form. Let D be the reduced diameter.

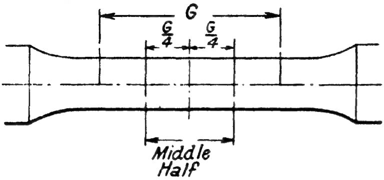


Fig. 5. Middle Half of Gauge Length of a Test Piece

$$\text{Reduction of area, per cent} = \frac{\frac{1}{4} - (\pi D^2/4)}{\frac{1}{4}} \times 100$$

$$= \frac{\frac{1}{4}\,(1 - \pi D^2)}{\frac{1}{4}} \times 100$$

$$= (1 - \pi D^2) \times 100$$

This obviates any division or calculation of area. For a closely approximate result the following empirical formula has the attraction of even greater simplicity.

$$\text{Reduction of area, per cent} = [(1 - 3D^2) \times 100] - 2$$

(for $\frac{1}{4}$ sq. in. cross-section)

A Tensile Test on Nickel-chrome Steel. This test is included to illustrate the observations made regarding the tensile test.

Fig. 7 (page 16) is a load/elongation diagram continued to just beyond the elastic line, and the table of extensions (page 11) is included in order to show the unfailing regularity with which the calculated extensions per ton per square inch are obtained during elastic extension.

The method of calculating the elastic modulus from the graph is also shown.

The cross-sectional area of the test piece being 0·25 sq. in. and the gauge being thereby 2 in., the curve could be shown on a stress/strain basis merely by multiplying the load scale by 4 and dividing the extension by 2. It is simply a matter of scale, without altering the graph in any way.

Test Bar Machined to 0·564 in. Diameter 2 in. Gauge Length. Note that as the bar has a cross-sectional area of 0·25 sq. in., 1-ton load increments are equivalent to a stress intensity of 4 tons/sq. in., and it was previously calculated that the extension due to such a stress intensity would be 0·00064 in., assuming a modulus value of 12,500 tons/sq. in. The 0·00064 in. is equal to nearly 13 divisions on the extensometer, each division of which equals 0·00005 in., and by the table of extensions (Table II, page 11) it is seen what a valuable check on results the calculated value can be.

For the calculation of the actual value of the elastic modulus the triangle ABC is drawn. AB represents the extension due to load BC, and, according to the scale to which the curve is drawn, $AB = 0{\cdot}0036$ in. $BC = 5{\cdot}6$ tons. Before the modulus can be calculated these values of extension and load must be converted into strain and stress respectively. Knowing that—

$$\text{Strain} = \frac{\text{extension}}{\text{gauge length}}$$

an extension of 0·0036 in. becomes a strain of 0·0018, and as stress = load/area, the stress intensity equivalent to a load of 5·6 tons = 22·4 tons/sq. in.

$$\text{Modulus} = \frac{\text{stress}}{\text{strain}} = \frac{22{\cdot}4}{0{\cdot}0018} = 12{,}450 \text{ tons/sq. in.}$$

TABLE II. TEST RESULTS FOR FIG. 7

Load	Ext. 1/20,000 in. on 2 in.	Dial divs. per ton load	Ext. per 4 tons/sq. in.
tons			in.
2	Dial set to 0	—	—
3	11	11	0·00055
4	23	12	0·00060
5	36	13	0·00065
6	49	13	0·00065
7	61	12	0·00060
8	74	13	0·00065
9	86	12	0·00060
10	99	13	0·00065
11	112	13	0·00065
11·5	120		
12	129	17	
12·2	133		
12·4	139		elastic limit exceeded
12·6	146	61	
12·8	160		
12·9	168		
13	190		

Any other triangle taken on the elastic line will give the same result, but the larger the triangle the less will be the error. The top of the curve does not indicate the point of maximum load, as the extensometer was removed after having obtained sufficient data, and this was before the maximum load had been reached.

The test figures are shown with the curve. For ascertaining the values of percentage elongation and reduction of area quickly and without calculation, two gauges are available. For elongation, the final gauge length is measured with dividers and this length transferred to a steel scale, when percentage elongation is read off direct. The scale is marked for numerous gauge lengths, and is illustrated in Fig. 6A. The reduction of area gauge is in the form of a tapered slot, the sides of which are ground to form knife-edges. The gauge is used to caliper the least diameter of the bar after fracture, and either the reduced diameter or the percentage reduction of area can be directly read from the scale on the blades of the gauge. Normally the gauge is made for one particular size of test bar, but the adjustable gauge shown in Fig. 6C is better suited when both standard and subsidiary standard test bars are in use. It also has the advantage of indicating the correct reduction of area per cent even if the test bar is inadvertently machined over or under size.

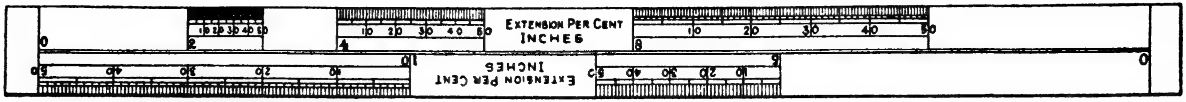


FIG. 6A. ELONGATION PER CENT GAUGE

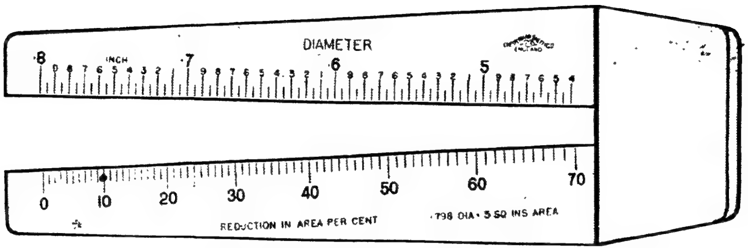


FIG. 6B. REDUCTION OF AREA PER CENT GAUGE

(Chesterman)

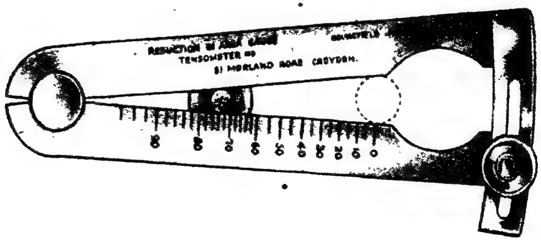


FIG. 6C. ADJUSTABLE GAUGE FOR REDUCTION OF AREA PER CENT

(Tensometer Ltd.)

In use the knurled screw is loosened and the loose arm adjusted until the zero graduation mark is on the diameter of the bar. The screw is then tightened and the gauge thus becomes set for that particular diameter.

SUMMARY OF DEFINITIONS

Elastic Limit or Limit of Proportionality is the stress at which a load extension or stress/strain diagram ceases to be a straight line, i.e. that stress at which strains cease to be proportional to the corresponding stresses.

Hooke's Law. This law states that up to the limit of proportionality, the stress divided by the corresponding strain is a constant. This constant is called the *elastic modulus*, or *modulus of elasticity* for the particular material, and is denoted by E—

$$\frac{\text{Stress}}{\text{Strain}} = \text{modulus of elasticity} = E$$

Yield Stress. This is defined as the lowest stress at which extension of the test piece increases without increase of load. Practically it is determined by certain indications given by the testing machine and checked by using dividers on the gauge length, to see that a visible permanent increase in gauge length has occurred. As many materials do not indicate a defined yield stress, the proof stress is used for such material.

Proof Stress. Proof stress is the stress necessary to cause a permanent extension, equal to a defined percentage of the gauge length.

Permanent Extension. Is that increase in length of the gauge length which remains when the load is removed. In order to accurately ascertain permanent extension it is advisable to subject the test piece to a tensioning load.

Maximum or Ultimate Stress. Is the maximum load divided by the original cross-sectional area of the gauge length of the test piece.

Percentage Elongation. This is expressed as the percentage increase in length of the gauge length, the distance between gauge points being measured after fracture.

If L = original gauge length and L_1 = final gauge length—

$$\text{Percentage elongation} = \frac{L_1 - L}{L} \times 100$$

Percentage Reduction of Area. Is the percentage decrease in cross-sectional area of the test bar, obtained by measuring the least diameter of the fractured specimen.

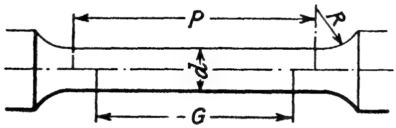
If D = least diameter—

$$\text{Percentage reduction of area} = (1 - \pi D^2) \times 100 \text{ for } 0{\cdot}25 \text{ sq. in. bar}$$

Gauge Length. Is the particular length of the parallel portion of the test piece over which extensions are measured. For cylindrical test pieces the gauge length used for calculation of percentage elongation is a function of the cross-sectional area or the diameter of the test bar.

$$\text{Gauge length in.} = 4 \times \sqrt{(\text{cross-sectional area sq. in.})}$$
$$\text{or} = 3{\cdot}54 \times \text{diameter in.}$$

TABLE III. STANDARD AND SUBSIDIARY STANDARD TENSILE TEST BARS



Diameter d	Cross-sect. A	Parallel length P	Gauge length G	Radius R	Remarks
in.	sq. in.	in.	in.	in.	
0·564	0·2500	2·250	2·00	0·5000	Standard bar
0·424	0·1412	1·750	1·50	0·3750	Subsidiary standards
0·399	0·1250	1·600	1·41	0·3750	
0·357	0·1000	1·500	1·26	0·3750	
0·282	0·0625	1·250	1·00	0·2500	
0·226	0·0400	1·000	0·80	0·2500	
0·159	0·0200	0·750	0·56	0·1875	
0·125	0·0122	0·625	0·44	0·1875	

The above dimensions refer to bar or forged material for aircraft purposes, the standard diameter of 0·564 in. being used wherever possible. For cast material the transition radius R is increased and for the standard bar is 3·5 in.

Test pieces cut from or representing small forgings or bars can be made to the subsidiary standard dimensions, but if a proof stress determination is to be made it will be necessary to increase the parallel length to 2·25 in. in order to attach the extensometer, which for aircraft material testing is usually made for a 2 in. gauge length. The final percentage elongation will of course be calculated on the particular gauge length given in Table III.

If it is necessary to increase the overall length of a test bar in order to suit the gripping arrangements of any particular machine, it is better to machine the extra length on the ends of the specimens than to increase the parallel length of the bar. If the parallel length is increased, fracture may occur outside the gauge length with consequent loss in percentage elongation. This is particularly so when testing cast test bars. As mentioned, however, for proof stress determinations on subsidiary standard test bars the parallel length must be increased. In these cases, and particularly when testing wire or rod, mark a succession of gauge lengths as given in Table III, at intervals of half a gauge length along the parallel length. The final extension is then measured on that gauge length in which the fracture occurs most centrally.

For a complete list of test bars reference should be made to British Standard Specification No. 18—1938.

CHAPTER II

PROOF STRESS DETERMINATIONS

PROOF stress is defined as the stress at which the stress/strain (or load/extension) curve departs by a specified percentage of the gauge length from the straight line of proportionality, i.e. the elastic line. As previously mentioned, many steels and non-ferrous alloys have no yield point as experienced with very mild steel. After the limit of proportionality is reached the extensions increase at a much faster rate, but there is no sudden large extension as denoted by the part of the curve C to D in Fig. 1 (page 1).

For design purposes, however, it is necessary to know some definite stress to which the material can be loaded with safety before any large permanent extension occurs, and in the absence of a definite yield some other point must be chosen.

If it were possible to determine the limit of proportionality readily in ordinary testing such a point could be chosen, but in order to accurately position the elastic limit, the precision instruments required and the time taken to plot the curve render such a point useless for ordinary commercial test requirements.

If, instead of trying to fix a minute point where the elastic line ceases to be straight and instead find a point where a definite amount of permanent extension has taken place, then the task becomes greatly simplified and readily adapted for commercial testing.

Such is the proof stress. It is simply that particular stress at which a definite amount of permanent extension has taken place. The amount of permanent extension allowable for any material is conveniently expressed as a percentage of the *extensometer* gauge length, and when testing material, reference has to be made to the relevant specification in order to ascertain the correct percentage. The percentages usually employed are 0·1 per cent, 0·2 per cent, 0·5 per cent. For example, if the gauge length is 2 in. and a 0·1 per cent proof stress is called for, it is the stress at which a permanent extension of 0·1 per cent of 2 in., i.e. 0·002 in., is obtained. The whole process really amounts to this: instead of looking for the needle in the haystack, look for the haystack instead, and then it is known that the needle must be within certain boundaries!

Method of Determining Proof Stress. Specifications which call for proof stress usually state that on one sample the proof stress shall be obtained from an accurately determined load/elongation diagram, and on remaining samples by an approved method. [*Note.* For load/elongation, stress/strain or stress/elongation can be substituted.]

The load/elongation diagram is the normal method of accurately determining the actual proof stress, and other methods are used chiefly to verify that the stress is either above a certain minimum limit or is between a minimum and a maximum limit, according to the requirements of the relevant specification. One method used gives a very close approximation

to the actual proof stress without the necessity for plotting a graph, and this will be dealt with in detail later.

Determination of the Actual Proof Stress from a Load/Elongation, Stress/Strain, or Stress/Elongation Diagram. Consider the load/elongation curve of Fig. 7. This may be taken as an accurately determined diagram for the purpose of ascertaining the proof stress. Sufficient points are taken

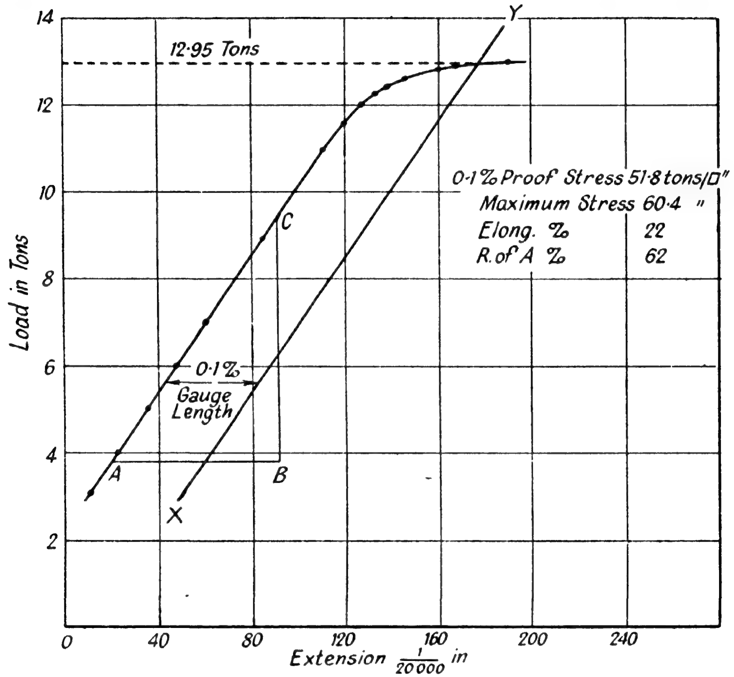


FIG. 7. LOAD/EXTENSION CURVE (NICKEL-CHROME STEEL)

on the elastic line to ensure that the line is truly straight and that both extensometer and testing machine are accurately functioning. After the elastic limit is passed, as denoted by the increasing extensions, the load increments are sufficiently reduced to enable sufficient points to be plotted in order to draw an accurate curve. This feature will be noted from Table II (page 11).

It will be noted that the diagram does not commence from zero load, and the reason for this may not be obvious. This is because readings of extension at light loads are not accurate, and the material needs a tensioning stress in order, as it were, to settle down. Therefore, in practice it is not customary to take heed of extensometer readings until a small initial load has been first applied. Having drawn the diagram to a suitable scale, the next operation is to determine the proof stress.

For the particular material under consideration, the specified amount of permanent extension allowed is 0·1 per cent of the gauge length, and the gauge length being 2 in., the amount will be 0·002 in. Now, what is required is to find that point on the curve where a deviation of 0·002 in. from the elastic line has occurred. This is simply done by drawing a line on the diagram parallel with the elastic line but 0·002 in. removed from it. This 0·002 in. is taken horizontally, not at right angles to the elastic line. The point where this line XY in Fig. 7 cuts the curve is therefore at that particular load which causes a departure of 0·002 in. from the elastic line, i.e. 0·002 permanent extension. To convert the load to stress, divide by the original cross-sectional area of the test bar, and that then is the 0·1 per cent proof stress.

If the elongation scale is on a strain basis, remember that an extension of 0·002 in. on a 2 in. gauge length is equal to a strain of 0·001, so that the parallel line XY would be drawn on the strain scale 0·001 removed from the elastic line.

Proof Stress not Less than a Minimum Value or Between Minimum and Maximum Limits. As it is obviously impossible for material to be manufactured, manipulated, and treated with absolute uniformity, specifications in which proof stress requirements are tabulated do not call for a definite value for the proof stress, but specify that it must be not less than some minimum value, or else be between fixed minimum and maximum limits. The load/elongation diagram method of obtaining the actual stress is used in order to discover with accuracy the relation that this stress bears to the limits.

Material from manufacturers which has satisfactorily passed specification requirements is, however, often subsequently manipulated and heat-treated in order to produce the finished product. Duralumin is a good example of this. If the finished product is for aircraft purposes, it is obviously essential to know that the material still conforms to specification requirements, so that test samples are always included in the final heat-treatment operation, and these are submitted for check testing. With these samples it is not usually necessary to plot a diagram, but simply to ascertain whether they conform to specification requirements in respect of the limits imposed. The methods employed for these tests are quickly carried out, and if it is required to know the approximate value of the actual stress, one particular method can be used which is accurate to less than 0·2 ton/sq. in.

In order to understand fully the particular methods employed it will be necessary to clarify certain other points relative to the proof stress curve.

Referring to Fig. 8A, $ADGH$ is a stress/extension curve, XY being drawn parallel with the elastic line AB and at a distance 0·002 in. from it. G represents the proof stress given by ordinate AE, and D the elastic limit denoted by ordinate AC. Now, a very popular misconception is that the elastic limit is the end of *all* elastic extension. It is believed that the extension which takes place after the elastic limit is all permanent. This is entirely incorrect, and is no doubt brought about by too great prominence being given to such a loose statement as "the elastic limit is the point where elastic extension ceases," and to the very term "elastic *limit*" itself. The expression *limit of proportionality* is far better and

more correct. The incorrect meaning assigned to the term "elastic limit" often leads engineers to think that the 0·002 in. permanent extension allowable for 0·1 per cent proof stress on a 2 in. gauge length is that stress where 0·002 in. *extension* beyond the elastic limit is reached. That this is not so will be clear from the explanation which follows.

After the limit of proportionality has been reached, the extension taking place is *partly elastic* and *partly permanent*. Fig. 8A shows that the total extension at the proof stress G is equal to EG, this being made up of EF and FG, F being the point where the horizontal line EG cuts the produced line of proportionality. Extension EF equals the *elastic* extension due to a stress AE (i.e. the proof stress), and FG represents the allowable permanent extension. The total extension at the proof stress is then seen to consist of—

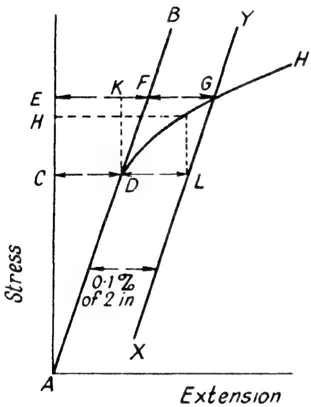


FIG. 8A. CURVE SHOWING EXTENSION AT PROOF STRESS

1. The elastic extension due to such a stress, and
2. The permanent extension allowed.

Note that this is entirely different from the extension at the limit of proportionality plus the permanent set, which in the figure is equal to CL this amount of extension occurring at stress H which obviously does not coincide with the proof stress E. It is seen that the amount of elastic extension between the limit of proportionality at stress AC and at the proof stress is equal to KF.

1. **Proof Stress not less than a Minimum Value.** Take, for example, a case where the specification states that the 0·1 per cent proof stress must not be less than 45 tons/sq. in., and the material for test is an alloy steel machined to a standard test bar form 0·564 in. diameter and 2 in. gauge length. Knowing the approximate value of the elastic modulus (12,500 tons/sq. in.), calculate the total extension which should occur at the minimum specified stress, less a certain amount of tensioning stress. This tensioning or "settling-down" stress may be 20–25 per cent of the specified minimum proof stress. The following formula will be useful for the necessary calculations—

Let stress $= S$ tons/sq. in.
elastic extension $= X$ in.
gauge length $= l$ in. (*extensometer gauge length*)
modulus $= E$

Then
$$\frac{\text{Stress}}{\text{Strain}} = E$$

or
$$\frac{Sl}{X} = E \qquad \left(\text{Strain} = \frac{X}{l}\right)$$

$$\therefore \quad X = Sl/E$$

$X =$ the elastic extension due to any stress S

If, therefore, it is required to ascertain the total extension at a proof stress of 45 tons/sq. in., to X must be added the permissible permanent extension allowed. In the case under consideration, the allowable permanent extension is 0·002 in., being 0·1 per cent of 2 in.

In order that readings may be accurate, a tensioning stress of 9 tons/sq. in. is taken and readings of extension up to that load neglected, i.e. the bezel of extensometer dial may be set to zero at 9 tons/sq. in. load on the bar. The total extension, then, as read off the dial, at a proof stress minimum of 45 tons/sq. in. (Fig. 8B), will be—

Total extension = (elastic extension at 45 tons/sq. in. — elastic extension at 9 tons/sq. in.) + the allowable permanent extension of 0·002 in.

= elastic extension due to a stress increase of 36 tons/sq. in. + 0·002 in.

i.e. $$X \text{ total} = \frac{36 \times 2}{12{,}500} + 0{\cdot}002 \text{ in.}$$

$$= 0{\cdot}00576 + 0{\cdot}002 \text{ in.}$$

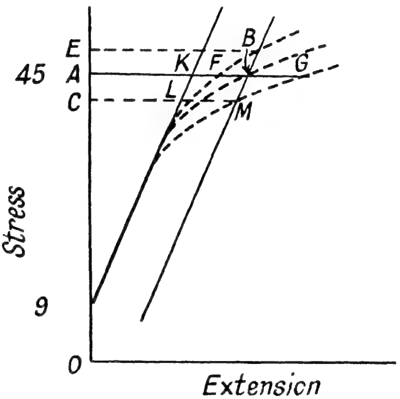


Fig. 8B. Proof Stress above a Minimum Limit

Now, if the material has a proof stress exactly on the minimum limit OA of 45 tons/sq. in., and the bar is first loaded 9 tons/sq. in. before readings are observed, the total extension indicated by the extensometer from 9 to 45 tons/sq. in. should be 0·00776 in., i.e. AB, the bezel of the dial having been set at zero at 9 tons/sq. in. If, for example, the proof stress was actually lower than 45 tons/sq. in., i.e. OC, then the 0·002 in. permanent extension LM would have occurred before a loading of 45 tons/sq. in. was obtained, and when the minimum limit of 45 tons/sq. in. was reached, the permanent extension KG would be in excess of 0·002 in. and the extensometer reading AG consequently *in excess of AB* or 0·00776 in.

If, on the other hand, the proof stress was actually higher than 45 tons/sq. in., e.g. OE, then at a loading of 45 tons/sq. in. the permanent extension KF would not have reached 0·002 in., and therefore the extensometer reading AF would be *less than* 0·00776 in.

These results, then, form the basis of the test, which is carried out as follows—

(*a*) Load the test piece to 9 tons/sq. in., i.e. 20 per cent of the minimum stated proof stress of 45 tons/sq. in.

(*b*) Set extensometer dial to zero.

(*c*) Load to 45 tons/sq. in.

(*d*) Read the extensometer immediately the load is reached. If the reading is less than 0·00776 in., then the proof stress is in excess of 45

tons/sq. in.; if the reading is just 0·00776 in. then the proof stress is 45 tons/sq. in. In these cases the material satisfies the specification requirements. If the dial reading is greater than 0·00776 in., then the proof stress is lower than 45 tons/sq. in. and the material fails to satisfy the specification requirements.

It will be observed that as the calculated elastic extension for 45tons/sq. in. is 0·00576 in., any variation from this figure when a stress of 45 tons/sq. in. is loaded, must be due to permanent extension.

Fig. 8B is for the purpose of explaining the method and has not to be drawn for the actual test.

[*Note.* As the stress is increasing from 9 tons/sq. in., check extension against load as outlined previously.]

2. **Proof Stress Between Minimum and Maximum Limits.** If, for example, the same steel as before had been specified to have a proof stress not less than 45 tons/sq. in. and not more than 55 tons/sq. in., then proceed precisely as before, but before loading calculate the total extension occurring at a proof stress of 55 tons/sq. in.

$$X \text{ total} = \frac{(55 - 9) \times 2}{12{,}500} + 0{\cdot}002$$

$$= 0{\cdot}00736 + 0{\cdot}002$$

$$= 0{\cdot}00936 \text{ in.}$$

Now, if the proof stress must lie between 45 and 55 tons/sq. in., it will most likely be between the two, i.e. above 45 but below 55 tons/sq. in. If the proof stress is below 55 tons/sq. in, then the 0·002 in. permanent extension will have occurred before that stress, so on reaching a stress of that amount more than 0·002 in. permanent extension will be present, thereby causing the calculated extension of 0·00936 in. to be exceeded.

After reaching operation (*d*) in the previous example, i.e. loading to 45 tons/sq. in., and ascertaining that the total extension is *not more than* 0·00776 in., continue loading until a stress of 55 tons/sq. in. is reached, when the extensometer should read *not less than* 0·00936 in.

The proof stress is then within the limits 45–55 tons/sq. in.

In practice the calculated extensions are converted to the equivalent number of extensometer dial divisions, but for easier explanation the extensions have been retained as calculated.

It will be noted that this method involves the assumption of an approximate value for the elastic modulus.

This, however, does not introduce errors of any consequence having regard to the method which is only to determine whether the proof stress is above a minimum limit, or between minimum and maximum limits. By plotting several load/extension diagrams for any one material, for example steel, to various specifications, the appropriate value for E can be taken instead of 12,500.

Although the procedure described above can be quickly carried out, there are two objections which can be raised against its general use (1) that a calculation based on an assumed value of the modulus has to be made, (2) that on material which has a low value of the proof stress,

loading to the specified minimum or maximum stresses may cause the extensometer pointer to advance so rapidly that no accurate readings are possible.

The second objection may be of greater account but for both reasons other methods are more often adopted to prove that a material is within the specified limits. These methods are explained subsequently.

3. **The Permanent Extension Method.** This method is in effect an adaptation of that just described, but readings of extension are based upon the permanent, and not the total extension. It also obviates the use of an approximate value for the elastic modulus, which may affect to a very slight degree the total extension. It must not be thought, however, that the error introduced by such an approximate value is of any consequence, as the plotting of a large number of load/elongation diagrams for any given material enables the approximation to be as accurate as necessary.

The chief value, perhaps, is that it dispenses with the need for any calculations of total extension to be made, but it is still necessary to know how such calculations are made. In order that the test may be accurately performed it is still necessary to check readings of elastic extension against increments of stress (e.g. 0·00016 in. elastic extension per ton per square inch per 2 in. gauge length for steel, as in Table II (page 11)), otherwise there is no check whatsoever on the test. Taking the same example as before, namely that of the steel which had the limits 45–55 tons/sq. in. for the proof stress, the following method is adopted.

It was previously explained that if the proof stress was actually below 45 tons/sq. in., then at 45 tons/sq. in. a permanent extension of more than 0·002 in. would be present, and if higher than 45 tons/sq. in., then at that stress there would not yet be 0·002 in. permanent extension.

Now, therefore, if the amount of permanent extension could be separated from the elastic extension, at the stress of 45 and of 55 tons/sq. in., it could be seen at a glance whether the material conformed with those limits. What, then, could be more simple than to remove the load? The elastic extension would disappear, and only the amount of permanent extension would be indicated by the extensometer. This, then, is the procedure—

1. Load the specimen until the tensioning stress is reached (about 20 per cent of the specified minimum proof stress).
2. Set the extensometer to zero (rotate the bezel).
3. Increase load for a small stress increment in order to check extensometer readings.
4. Load until the minimum proof stress is reached (checking extensions against further stress increments, if required).
5. Hold the load at the minimum proof stress for 15 sec.
6. Unload to just below the tensioning stress.
7. Reload to the tensioning stress.
8. Read the extensometer.

This will indicate the amount of permanent extension and should not exceed the amount which is permitted by the relevant specification. The actual amount will naturally depend upon the percentage proof stress (i.e. 0·1 per cent, 0·2 per cent, etc.) and the particular gauge length used.

Taking the steel specimen as an example, 0·1 per cent proof stress and 2 in. gauge length, then the amount of permanent extension shown should not exceed 0·002 in.

The above procedure is all that would be necessary if only a minimum limit were specified. If a maximum limit, as for example the 55 tons/sq. in., were specified in addition, and having seen that the material was satisfactory in respect of the minimum proof stress, repeat operations 4–8 only in place of minimum proof stress, substitute maximum proof stress.

[*Note.* Do not set the extensometer to zero before again proceeding with operation 4.] If the proof stress is actually lower than the 55 tons/sq. in., when such a stress is reached *more* than 0·002 in. permanent extension will be present, and this will be read on the dial at operation 8. Thus if the permanent extension in the first case was not more than 0·002 in. and in the second case not less than 0·002 in., then the proof stress must be between 45 and 55 tons/sq. in. as required.

It will be noted that for operation 6 it is necessary to work in a reverse direction both the tensile-testing machine and the extensometer, i.e. with decreasing loads and extensions. This is liable to introduce small errors into the results unless the machine and extensometer are very sensitive. In this respect it is laid down that the extensometer should possess a sensitivity of 0·00005 in., and that the reading at any given load on loading should not differ from that obtained at the same load on unloading by more than 0·00005 in. per inch of gauge length.

Operation 6 (in which unloading is carried below the datum line of the test) and operation 7 (again loading up to the datum line) are for the purpose of minimizing any errors. These methods of determining proof stress above a minimum limit or within a range do not indicate precisely where the proof stress actually is. If a method could be evolved which, in addition to indicating that the proof stress was within a range, also indicated with fair precision what the actual stress was, an obvious advantage would accrue. The information obtained from the test could become more definite and enable the uniformity of heat treatment or processing to be closely watched. If, for example, finally heat-treated duralumin test strips consistently gave proof stress figures of, say, 16·5 tons/sq. in. and the figure suddenly dropped to 15·5 tons/sq. in. or below, then obviously some unsatisfactory feature regarding the heat treatment would be made apparent and enable the operator to seek a cause. Although the 15·5 tons/sq. in. would pass the specification figure for duralumin, yet the detection of the cause of the drop would probably save subsequent material from giving a proof stress even lower than the required minimum. Such a test is available, and it will be dealt with now.

4. **The Four-point Method of Determining Proof Stress.** This very ingenious method, evolved by N. Lindley of the Aeronautical Inspection Department of the Ministry of Aircraft Production, in conjunction with C. Crow of Firth-Vickers Stainless Steels, Ltd., will be explained in detail, the various calculations regarding the determination of proof stress having now been sufficiently considered to enable the method to be fully understood and its value to be appreciated. For testing "released" aircraft material which has been subsequently heat-treated and for all routine

testing to B.S. specification requirements where proof stress is required, this method is decidedly superior. In order to obtain a closely approximate value of the proof stress by this method it is necessary that a lower and an upper limit of the proof stress be specified. In many cases only a lower limit is specified, but this need be no deterrent, as an upper limit can be assumed, this upper limit being based on a knowledge of the material obtained from practical testing experience.

The four-point system can be used in preference to the permanent extension method to show that a material is satisfactory in respect of proof stress above a minimum value or between limits, even if it is not desired to know the closely approximate value of the actual stress. The advantage of the four-point method is that no unloading of the testing machine or extensometer is necessary.

If the calculations under the headings of proof stress determinations and the other considerations enumerated were understood, no difficulty should be experienced in understanding the principles of the four-point method. Although in the practical application of the method no diagram need be plotted, yet a diagram is necessary in order to understand fully the working of the test. Referring to page 19, it was shown that the total extension at any particular proof stress could be easily calculated from the amount of elastic extension and the permanent extension allowable.

In the four-point method, instead of assuming an approximate value for the elastic modulus, the elastic extension is calculated from the extensometer reading for a definite stress increment when the material is under test. Referring to Fig. 9, $ABCD$ is a stress/elongation diagram, in which XY is drawn parallel to the line of proportionality and px represents the specified permanent extension allowed. OA represents the tensioning stress, OL and OU being the lower and upper limits of the specified proof stress range. AF and FG are equal increments of stress for the purpose of observing extension readings x_1 and x_2.

As it is often more easy to follow examples in which actual figures are used instead of symbols, it will be assumed that Fig. 9 represents the working of the four-point method as applied to duralumin sheet solution treated by a sub-contractor. For this material the minimum specified stress will be as taken 15 ton/sq. in. (in some cases it is 14·5 tons/sq. in.) and the chosen upper limit 18 tons/sq. in. The upper limit is set from a knowledge of the material and its treatment. As 0·1 per cent proof stress is specified for this material and a gauge length of 2 in. is employed, px will be 0·002 in. The first operation is to apply the tensioning stress OA, which can be 20 per cent of the specified minimum proof stress OL, 15 tons/sq. in. Apply, therefore, a tensioning stress OA of 3 tons/sq. in. and set the extensometer to zero.

Next, the first increment of stress AF is to be applied, and to make subsequent calculations more easy the value of AF should be chosen so that it is divisible a whole number of times into both AL (which is $OL - OA = 15 - 3 = 12$) and AU (which is $OU - OA = 18 - 3 = 15$). AF, therefore, can also be 3 tons/sq. in. Increase then the load from the datum line OA a further 3 tons/sq. in. to OF, and read extension x_1.

Now, from a knowledge of the elastic modulus of duralumin, the elastic extension x_1 due to a stress intensity of 3 tons/sq. in. can easily

be calculated, and should be done before the actual test commences. The modulus varies from 10·0 to 10·8 × 10^6 lb./sq. in., so that the average extension is calculated thus—

$$x_1 = \frac{S \times l}{E}$$
$$= \frac{3 \times 2240 \times 2}{10{,}400{,}000}$$
$$= 0{\cdot}00129 \text{ in.}$$

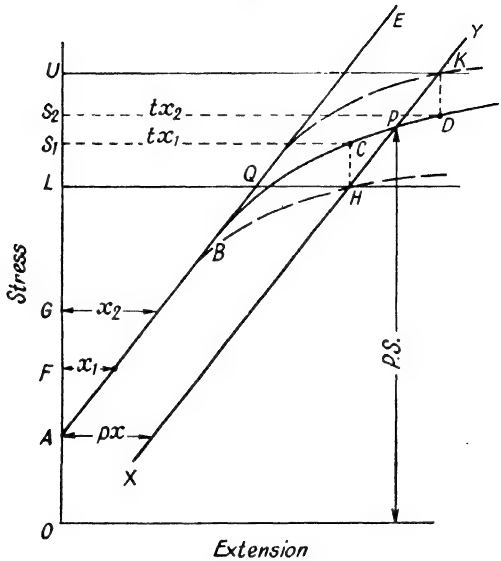


FIG. 9. FOUR-POINT METHOD OF PROOF STRESS DETERMINATION (1)

This reading on a dial indicating 1/20,000 in extension per division will equal nearly 26 divisions. Therefore at stress OF, and setting the dial to zero at OA, approximately 26 ± 1 divisions should show on the dial.

Having observed that this is so, it can be taken that the test is proceeding satisfactorily, but to make doubly sure another increment of 3 tons/sq. in. could be loaded, i.e. to OG, and the reading x_2 should be double that of x_1 ± 0·00005 in. (± one dial division). It will be seen that so long as the material is extending elastically, equal stress increments will simply multiply the original extension x_1.

Now $$\frac{x_1}{AF} = \frac{LQ}{AL} \text{ or } LQ = \frac{AL}{AF} \times x_1$$

In words this means, the elastic extension LQ due to a stress AL will be in the same proportion as the elastic extension x_1 is to the stress AF. For example, if a stress of 3 tons/sq. in. produces an elastic extension of 27

dial divisions, then a stress of 12 tons/sq. in. will produce an elastic extension of $\frac{12}{3} \times 27$ divisions.

Remembering that the total extension at a proof stress of 15 tons/sq. in. will be made up of the elastic extension at that stress plus the allowable permanent extension, then the total extension at a proof stress of 15 tons/sq. in. will be—

$$LH = LQ + QH$$

$$\text{as } LQ = \frac{AL}{AF} \times x_1 \text{ and } QH = px$$

$$LH = \left(\frac{AL}{AF} \times x_1\right) + px = \left(\frac{12}{3} \times 27\right) + 40^*$$

$$= 148 \text{ divs.}$$

Let tx_1 represent this total extension at a proof stress of 15 tons/sq. in. Instead of plotting a number of equal points on the curve, the total extension is thus calculated from the first reading x_1.

The same calculation is then made on the assumption that 18 tons/sq. in. is the proof stress.

Total extension at a proof stress of 18 tons/sq. in. OU—

$$UK = \left(\frac{AU}{AF} \times x_1\right) + px$$

$$= \left(\frac{15}{3} \times 27\right) + 40$$

$$= 175 \text{ divisions}$$

That is, if the material has a proof stress of 15 tons./sq. in., and assuming $x_1 = 27/20{,}000$ in., the total extension measured from A should equal 148 dial divisions; if of 18 tons/sq. in., the extension will be 175 divisions. Let tx_2 represent this total extension at a proof stress of 18 tons/sq. in.

Now, as previously explained under methods 1 and 2, if the proof stress is *above* 15 tons/sq. in., then the extension of 148 divisions will not be obtained until some higher stress is reached; also, if *below* 18 tons/sq. in., 175 divisions will be registered before that latter stress. Calculations tx_1 and tx_2 are made, then, as soon as extension x_1 is recorded, and the next step is to increase the load until extension tx_1 is observed, noting the particular load at which the extension occurs. Again increase the load until extension tx_2 is observed, and note this load. Convert the loads at tx_1 and tx_2 to stress intensities which will give stresses S_1 and S_2, corresponding to points C and D on the curve.

At stress S_1, therefore, there is an extension of 148 dial divisions, calculated on the basis of a proof stress of 15 tons/sq. in., the 148 being made up of 108 divisions elastic extension plus 40 divisions (0·002 in.) permanent extension.

If S_1 is greater than 15 tons/sq. in. there must therefore be *more* elastic extension at S_1 than at 15 tons/sq. in.; consequently if in the fixed total

* Forty dial divisions = 0·002 in.

of 148 divisions more than 108 are due to elastic extension, then there must be less than 40—i.e. *less* than 0·002 in. permanent extension at S_1.

Similarly, if stress S_2 is lower than 18 tons/sq. in., then at S_2 (135 + 40), there must be less elastic extension than at 18 tons/sq. in., i.e. less than 135 dial divisions. Thus if less than 135 divisions represent the elastic extension, there must be more than 40 divisions, i.e. *more* than 0·002 in. permanent extension at S_2. These features can be noted from Fig. 9.

From C draw a line parallel with the stress ordinate until it meets the horizontal line through L, representing stress OL, i.e. the lower limit of 15 tons/sq. in. at H.

Now, had 15 tons/sq. in. been the actual proof stress of the material, tx_1 would have been at H, and therefore XY would have cut the stress/elongation curve at that point as shown by the lower dotted curve.

From point D draw a line parallel with the stress ordinate until it meets the horizontal line representing the upper limit of proof stress OU at K. As before, had the material actually possessed a proof stress of 18 tons/sq. in., the extension tx_2 would have been observed at stress OU, and XY would have cut the curve at K as shown by the upper dotted curve.

Note very carefully, then, these significant features.

(*a*) At stress S_1 there is less than 0·002 in. permanent extension, and at stress S_2 there is more than 0·002 in. permanent extension.

(*b*) Having obtained stresses S_1 and S_2 corresponding to extensions tx_1, tx_2, points H and K can be fixed, and by joining these the line XY can be drawn *without any other data whatsoever*.

(*c*) That over the length CD the curve is approximately a straight line.

(*d*) The actual proof stress P is between points C and D (where XY cuts the curve).

That part of the curve between the lower and upper limit of proof stress is reproduced in Fig. 10, to which reference should now be made. In this figure curve CD is replaced by a straight line joining those two points, and XY is obtained as in (*b*).

Let d = the proof stress range,

a = the amount of stress that S_1 is above the lower limit,

b = the amount of stress that S_2 is below the upper limit,

c = the amount of stress that the actual proof stress is above the lower limit.

Draw MN parallel with the stress ordinate through the actual proof stress P.

Then, by similar triangles HCP, KDP,

$$HC : KD = HP : KP$$

and from triangles HNP, KMP—

$$NP : MP = HP : KP$$

i.e.
$$HC : KD = NP : MP$$

or
$$a/b = NP/MP$$

Now,
$$NP = c$$
$$MP = d - c$$
i.e.
$$a/b = c/(d - c)$$
$$a\,(d - c) = bc$$
$$ad = bc + ac = c\,(a + b)$$
$$\therefore \quad c = ad/(a + b)$$

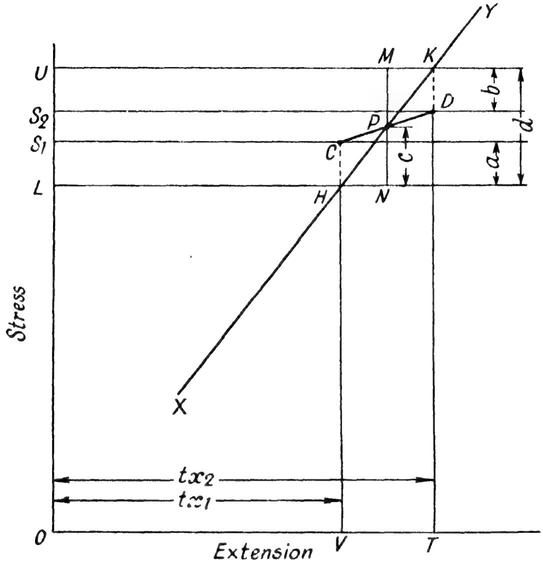


FIG. 10. FOUR-POINT METHOD OF PROOF STRESS DETERMINATION (2)

Therefore the actual proof stress is equal to the lower limit of the proof stress range OL, $+ c$.

By taking CD as a straight line the error does not amount to more than about 0·1 ton/sq. in.

Although the explanation of the four-point method seems involved, the practical application is very simple, as no graph need be drawn. The following example is based on such a test applied to solution treated duralumin sheet. Proof stress range, 15–18 tons/sq. in.; cross-section of test specimen, 0·028 in. (22 S.W.G.) × 0·500 in. wide, gauge length 2 in.; cross-sectional area, 0·014 sq. in.

Load for a tensioning stress of 3 tons/sq. in.:
$$\frac{\text{Load}}{\text{Area}} = \text{stress}$$
$$\therefore \quad \text{Load} = \text{area} \times \text{stress}$$
$$= 0{\cdot}014 \times 3$$
$$= 0{\cdot}042 \text{ ton}$$

Tensioning load = 0·042 ton

Tensioning load + an increment of 3 tons/sq. in. = 0·084 ton.

Procedure

(*a*) Load to 0·042 ton and set extensometer to zero.

(*b*) Increase load to 0·084 ton and read extension. (From previous calculation this should be about twenty-seven 1/20,000 divisions.)

Actual extension observed = 26 divisions.

(*c*) Increase load until extensometer reads—

$$\left(\frac{15-3}{3} \times 26\right) + 40 = \left(\frac{12}{3} \times 26\right) + 40 = 144 \text{ divisions}$$

and note the load.

Actual load was 0·226 ton.

(*d*) Increase load until extensometer reads—

$$\left(\frac{18-3}{3} \times 26\right) + 40 = \left(\frac{15}{3} \times 26\right) + 40 = 170 \text{ divisions}$$

and note the load.

Actual load was 0·239 ton.

(*e*) Remove extensometer and continue loading to ascertain the maximum load.

Actual maximum load was 0·382 ton. Percentage extension = 18.

Then $$S_1 = 0{\cdot}226/0{\cdot}014 = 16{\cdot}14 \text{ tons/sq. in.}$$

$$S_2 = 0{\cdot}239/0{\cdot}014 = 17{\cdot}07 \text{ tons/sq. in.}$$

This shows that the material is between 15 and 18 tons/sq. in. as specified.

$$a = 1{\cdot}14,\ b = 0{\cdot}93,\ d = 3$$

$$c = \frac{1{\cdot}14 \times 3}{1{\cdot}14 + 0{\cdot}93} = 1{\cdot}65 \text{ tons/sq. in.}$$

$$\therefore \quad \text{Proof stress} = 15 + 1{\cdot}65 = 16{\cdot}65 \text{ tons/sq. in.}$$

$$\text{Maximum stress} = 0{\cdot}382/0{\cdot}014 = 27{\cdot}29 \text{ tons/sq. in.}$$

As an example, it is advised to plot Fig. 10 (page 27) from the data of the above test, and, omitting MN, find the stress at which XY cuts CD. It should be at the calculated proof stress of 16·65 tons/sq. in.

Although in the above example the calculation of the total extensions has been set out in detail, there is no need for any calculation when normally using the method. By the preparation of a simple table, the total extensions for any proof stress limits can be immediately ascertained when the value of x_1 is obtained. An example of such a table is given opposite. (Table IV).

Determination of the Approximate Proof Stress if below the Minimum Specified. A knowledge of the principles involved in the determination of proof stress by the four-point method enables the stress to be estimated should it be below the specified minimum. Although the actual proof stress of duralumin sheet $L3$, which has been finally heat-treated for processing is usually between 15 and 18 tons/sq. in., yet cases do occur

TABLE IV. ALUMINIUM ALLOYS

(2 in gauge length, extensometer reading 1/20,000 in.)

X_E Dial divs. per 3 tons/sq. in.	X_T at 0·1 per cent proof stress	
	15	18
24	136	160
25	140	165
26	144	170
27	148	175

X_E = elastic extension.
X_T = total extension (i.e. elastic + permanent).

where due to faulty heat treatment the figure does fall below the minimum value. If, therefore, a four-point test on duralumin sheet is being carried out as the previous example, and the stresses S_1 and S_2 work out to be 14·3 and 15·1 tons/sq. in. respectively, how then is the closely approximate proof stress to be determined? In this case it cannot be said that the actual stress is between 14·3 and 15·1 tons/sq. in., as a little consideration will show.

The stress S_1 is based upon the total extension beyond the tensioning stress, due to a proof stress of 15 tons/sq. in. If, then, this amount of extension is obtained at a stress of 14·3 tons/sq. in. it means that *more than* 0·002 in. permissible permanent extension has occurred at that stress. This is because the elastic extension due to a stress of 14·3 tons/sq. in. will be less than that for a stress of 15 tons/sq. in., and therefore for a *given amount of total extension* (which equals the elastic extension plus the permanent extension allowed), if the amount of elastic extension is reduced, the amount of permanent extension must be increased. This will be more clear if another diagram is drawn.

In Fig. 11 the extensions tx_1 and tx_2 are marked at the corresponding stresses S_1 and S_2.*

$$tx_1 = 144 \text{ divisions}$$
$$S_1 = 14{\cdot}3 \text{ tons/sq. in.}$$
$$tx_2 = 170 \text{ divisions}$$
$$S_2 = 15{\cdot}1 \text{ tons/sq. in.}$$

Construct the diagram in accordance with Fig. 10 (page 27).

Now, any part of the curve to the right of XY (which represents the limit of specified permissible permanent extension) will be in excess of the 0·002 in. allowed. Further, XY does not cut the curve between C and D as previously, but by producing DC until it cuts XY, the approximate actual proof stress is obtained. From the diagram it is seen to be at 14·05 tons/sq. in., i.e. lower than stress S_1.

Having graphically obtained the approximate value of the proof

* The figures given are the results of an actual test.

stress, and showing that it is less than S_1, calculate the value of c in the expression $c = ad/(a + b)$ by treating a as a negative quantity, i.e.—

$$c = \frac{-0{\cdot}7 \times 3}{-0{\cdot}7 + 2{\cdot}9}$$

$$= \frac{-2{\cdot}1}{2{\cdot}2} = -0{\cdot}95$$

$$\text{Proof stress} = 15 - 0{\cdot}95$$

$$= 14{\cdot}05 \text{ tons/sq. in.}$$

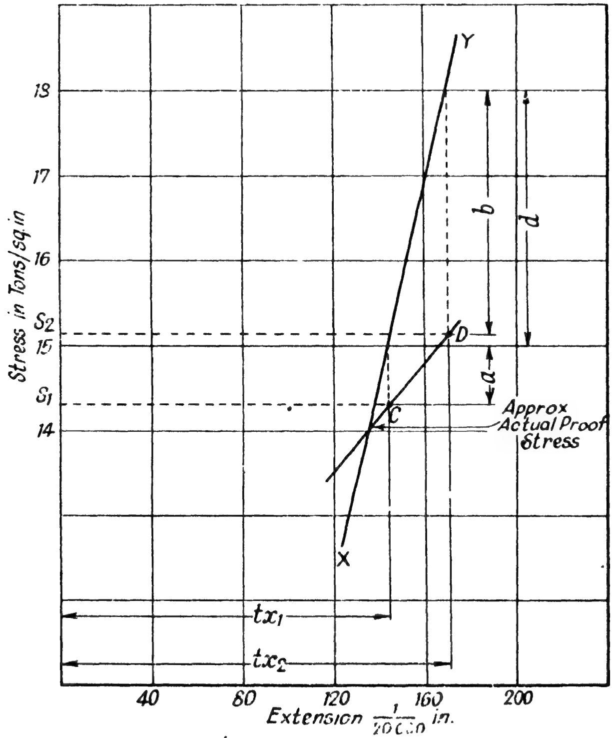


FIG. 11. PROOF STRESS BELOW A MINIMUM LIMIT

It is seen that if S_1 is below the minimum value, the proof stress can be calculated as before without the drawing of a graph.

If a low value of the proof is suspected, it is better to base a reading on the assumption of 12 tons/sq. in. as well as 15 and 18 tons/sq. in. Then if S_1 and S_2 both fall between 12 and 15 tons/sq. in., the proof stress may be calculated by treating those stresses as the upper and lower limits.

In order to demonstrate that the closely approximate proof stress is still obtainable by treating a as a negative quantity, several tests have

been carried out on duralumin sheet (Spec. *L*3) of low proof stress and readings taken based on 12, 15 and 18 tons/sq. in. The proof stress has then been calculated in the normal manner by taking the limits of 12 and 15 tons/sq. in., between which stresses S_1 and S_2 were located. It was then assumed that the operator was not expecting a low value for the proof stress and in consequence had only taken readings based on 15 and 18 tons/sq. in. proof stress. The very close agreement in the results of such a test are evident by the following example.

Test piece. Duralumin, Spec. 5L3, 0·037 × 0·5 in.
Cross-section area 0·0185 sq. in.
Extensometer set to zero at 0·0555 tons load, i.e. 3 tons/sq. in.
Reading at 6 tons/sq. in. = 26 divisions.
,, 9 ,, ,, = 52 ,,

Basis of proof stress	Total extn.	Load	Stress
tons/sq. in.	divs.	tons	tons/sq. in.
12	118	0·260	14·06
15	144	0·272	14·70
18	170	0·278	15·03

(1) Taking limits of 12 and 15 tons/sq. in.

$$c = \frac{ad}{a+b} = \frac{2{\cdot}06 \times 3}{2{\cdot}06 + 0{\cdot}3} = 2{\cdot}62$$

Proof stress = 12 + 2·62 = 14·62 tons/sq. in.

(2) Taking limits of 15 and 18 tons/sq. in.

$$c = \frac{-ad}{b-a} = -\frac{0{\cdot}3 \times 3}{2{\cdot}97 - 0{\cdot}3} = -0{\cdot}337$$

Proof stress = 15 − 0·337 = 14·663 tons/sq. in.

Therefore to the first decimal place, which suffices for any proof stress, the difference between the results is negligible.

This fact is of great assistance when conducting a test on a transversely cut sheet test piece on which the minimum allowable 0·1 per cent proof stress is 14·5 tons/sq. in. (Spec. L3). The limits 15, 18 tons/sq. in. can still be taken with confidence as the correct proof stress can be estimated even if S_1 is below the chosen limit of 15.

Proof Stress above a Maximum Limit. If, for example, in the case of duralumin, the upper limit of 18 tons/sq. in. were chosen as the arbitrary maximum, and calculations of total extension made, as before, on the basis of the proof stress occurring at 15 tons/sq. in. and at 18 tons/sq. in., and if on loading the test piece until extension tx_2 were obtained, the stress S_2 were in excess of 18 tons/sq. in., then the proof stress would actually be higher than S_2.

The reason for this is that the total extension being equal to the sum of the elastic extension at 18 tons/sq. in. and the 0·002 in. permanent extension allowed (minus the tensioning stress extension), if that total amount is not obtained during the test until a stress in excess of the 18 tons/sq. in. is reached, then the amount of *elastic* extension at S_2 will

be greater than that at 18 tons/sq. in. Consequently, if in the *fixed* total extension tx_2 the elastic part is increased, then the permanent part must be reduced, or in other words there is not yet 0·002 in. permanent extension at S_2.

If these circumstances arise, how can the proof stress be closely approximated as in the four-point method? As before, plot the results obtained from the test in the manner of Fig. 11, then, having obtained a graphical proof, the mathematical solution can be obtained.

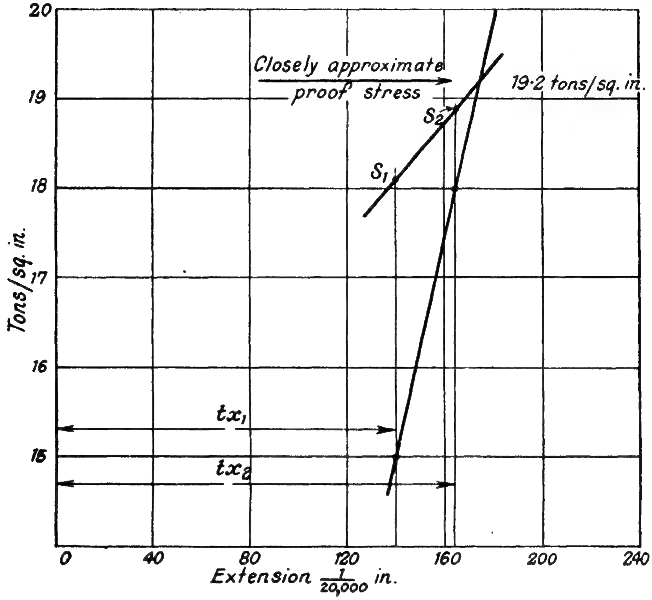


FIG. 12. PROOF STRESS ABOVE A MAXIMUM LIMIT

Fig. 12 shows the actual test results, in which it is seen that both stresses S_1 and S_2, corresponding to extensions tx_1 and tx_2, occur above 18 tons/sq. in.

Stress at tx_1 (140 dial divs.) = 18·1 tons/sq. in.

Stress at tx_2 (165 dial divs.) = 18·9 tons/sq. in.

By similar construction to that employed in Fig. 11, it is seen that the closely approximate value of the proof stress is at 19·2 tons/sq. in.

Applying the formula $c = ad/(a + b)$, where $a = 3{\cdot}1$, $d = 3$, and $b = -0{\cdot}9$, as it is above 18 tons/sq. in.—

$$c = \frac{3{\cdot}1 \times 3}{3{\cdot}1 - 0{\cdot}9}$$
$$= 4{\cdot}228$$

Approximate proof stress = 15 + 4·228 tons/sq. in.
= 19·2 tons/sq. in.

It will be noted therefore that whether the proof stress comes above or below the limits chosen, the actual stress can be determined with close approximation, and this fact should give confidence to any operator using the method as the result is safeguarded even if S_1 falls below the lower limit or S_2 above the upper limit. It is impossible to make mistakes as even if the cross-sectional area of a sheet test piece be incorrectly calculated the error will be evident after applying the stress increment, as the

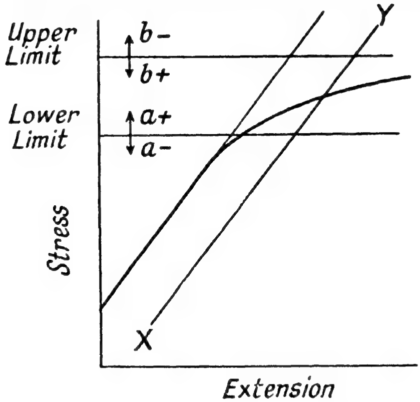


FIG. 13. FOUR-POINT METHOD, VALUES FOR a AND b

extensometer will read plus or minus the standard extension to a much greater degree than permissible.

In the author's opinion this method of determination is the most ingenious routine test yet devised, and demonstrates in a practical manner the use to which a knowledge of elastic and permanent extension can be put. There is no question as to the accuracy of the method when S_1 and S_2 fall within the allotted limits, as the result it gives will be within the same limits as those given by load or stress-elongation diagrams on the same uniform material, but tested on different machines or the same machines in different test houses. It must be realized that testing machines cannot be made as sensitive as chemical balances, and that the load indications cannot be read with the precision of a slide rule. These facts are of considerable importance in cases when the extensometer starts "running away" when registering extensions on the curved portion of a graph—the all important portion for proof stress determination by a plotted graph. Often far greater accuracy is achieved by taking loads only at the given extensions of the four-point method than trying to read loads and extensions simultaneously for the plotting of a graph.

The examples worked refer only to duralumin, a 2 in. gauge length, a 0·1 per cent proof stress, and an extensometer having each division equal to 0·00005 in. Sufficient detail has been given, however, to enable the proof stress to be calculated for any material. The procedure is exactly similar, only using the particular values of stress and extension applicable to the material in question. The test is greatly simplified

by using the number of dial divisions instead of the actual extension in inches.

Proof Stress Indicator for Four-point Method. When a number of proof stress determinations are made by means of the four-point method the solution of the essential formula

$$c = \frac{ad}{a + b}$$

becomes very tiresome, notwithstanding the rapidity of the test.

In order to dispense with the need for solving the formula directly, an indicator has been devised by means of which the proof stress can be ascertained without calculation within about five seconds from obtaining the values of stresses S_1 and S_2.

Referring to Fig. 10 (page 27) and its constructional explanation, and also to Table IV, it will be noted that the difference between extension tx_1 and tx_2 for 3 tons/sq. in. will be according to the initial basic extension X_1 and for the aluminium alloy will be either 24, 25, 26 or 27 dial divisions. In addition, from Fig. 10, it is evident that the slope of XY will also vary according to this difference.

With these considerations the special chart, Fig. 14, is constructed to embrace the possible variations in extension, and the slope of XY for the range of 3 tons/sq. in. stress and the stress scale is transferred to the ordinate CV and DT of Fig. 10.

If, then, a test had been performed as given on pages 27 and 28 the values of S_1 and S_2 could be located on their respective scales, and the point of intersection of a straight line drawn between these stresses and the appropriate slope XY would immediately indicate the proof stress as denoted in Fig. 14.

A transparent rule with a hair line would, of course, obviate the need for drawing a line between stresses S_1 and S_2, but there is a difficulty introduced when it is attempted to first locate the line on S_1, then move the other end of the rule to locate S_2, as the rule will move away from S_1. In order to overcome this disadvantage a mechanical movement was designed which maintained the location of S_1 whatever position was needed for S_2, and the complete instrument is illustrated in Fig. 15.

The cursor carries an extended arm on which is marked a central reading line. A hole is drilled on this line so that its axis is on the stress ordinate for scale S_1, this hole forming the pivot of the main rule.

The reading lines of the cursor arm and rule intersect at the axis of the pivot.

Immediately the two stresses S_1 and S_2 are known, the cursor is moved until the reading line of the arm is located on stress S_1, this movement effectively locating the axis of the pivot on that stress. The free end of the rule is then moved until its reading line is located at stress S_2, on the particular vertical line corresponding to extension X obtained during the initial part of the four-point test. The proof stress is then read directly from the corresponding slope of XY.

No errors are possible because no mathematical calculations have to be made, and the device requires no skill to operate.

The Three-point Method. For some routine tests when it is only neces-

sary to determine that a material has a proof stress above a certain specified minimum value, a variation of the four-point method is adopted.

This method, known as the three-point method, does not enable a closely approximate value of the proof stress to be obtained unless the value of stress given by the method is close to the minimum limit.

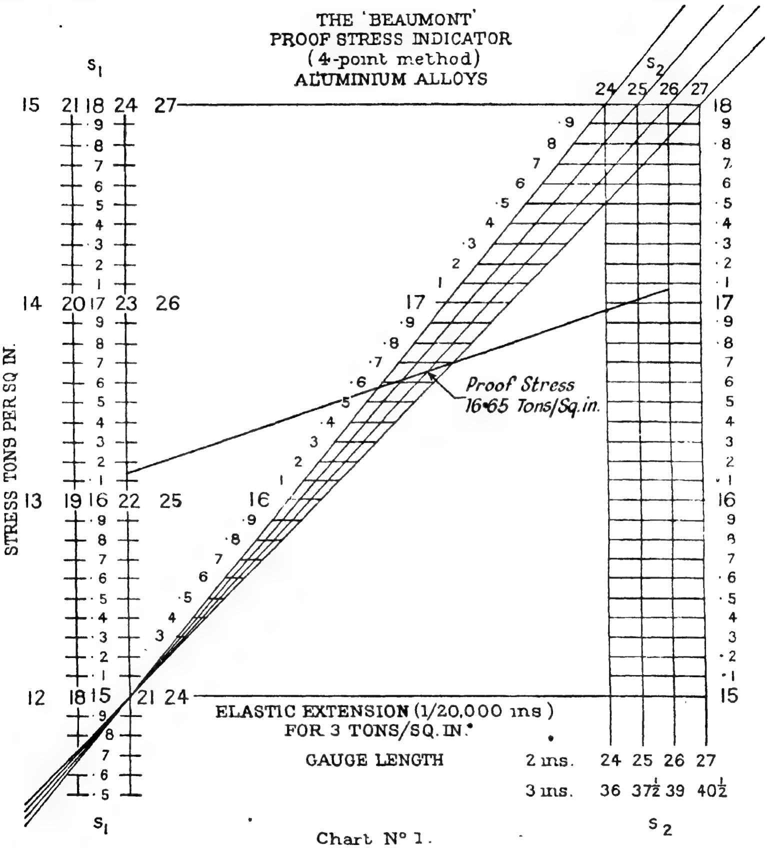


FIG. 14. CHART FOR THE "BEAUMONT" PROOF STRESS INDICATOR

The operation of the test is precisely the same as for the four-point method up to the stress S_1 (which is the third point). If stress S_1 is at, or above, the minimum specified, then the material is satisfactory in respect of the specification requirement.

The only advantage of this method is that the load (for stress S_2), at the second extension, has not to be noted and there is this very small saving in time. As however this is obtained at the expense of being able to ascertain the actual proof stress, it must be a doubtful advantage.

It is obviously an advantage to know with precision what the actual proof stress is, as the heat-treatment temperatures can then be closely controlled to give the required value.

If the three-point method is used for some applications it must be realized that the value of S_1 cannot be definitely related to the actual proof stress value. For example, if the value of S_1 was 16·0 tons/sq. in. in one test and 16·2 tons/sq. in. in another, the immediate likely conclusion would be that the second material would have a higher actual proof stress. This conclusion has little foundation, as if four-point readings are taken

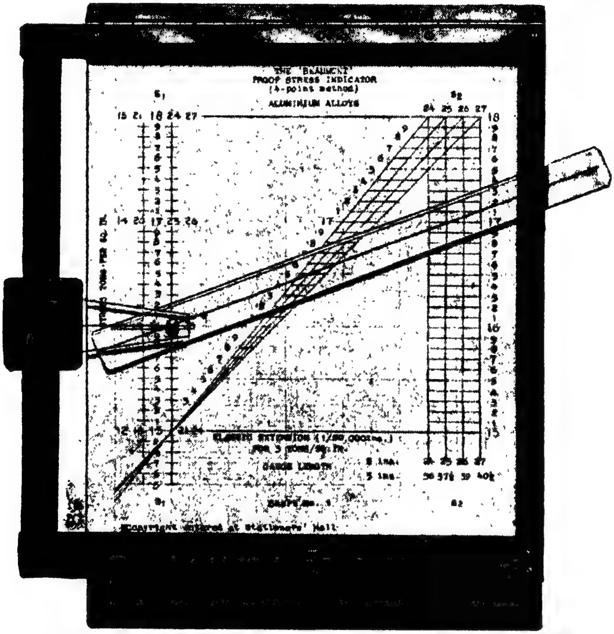


FIG. 15. THE "BEAUMONT" PROOF STRESS INDICATOR
(*J. E. Baty & Co. Ltd.*)

it may quite well be found that the proof stress is the same in each case as the following readings illustrate—

S_1	S_2	Proof stress
16	17	16·5
16·2	16·8	16·5

The solution of the formula

$$c = \frac{ad}{a + b}$$

has been an objection to the use of the four-point method when time is of great importance, but with the advent of the Proof Stress Indicator previously described, there seems little reason why the three-point should be used in preference to the more definite four-point method.

THE ESTIMATION OF PROOF STRESS FROM LOAD/ELONGATION TEST RESULTS

The knowledge that the total extension at the proof stress equals the amount of elastic extension at that stress plus the permissible permanent extension enables a quick estimation of proof stress to be made from test results.

Referring to Table V (below) it will be noted that the elastic extension per 0·02 ton load increment is 3 dial divisions, i.e. 3/20,000 in. While the material is extending along the line of proportionality, each 0·02 ton load increment will produce this extension, but when the limit of proportionality has been exceeded, a greater extension will occur. The extension in excess of the 3 divisions represents the amount of permanent extension per 0·02 ton load. Therefore when the amount of permanent extension totals 0·002 in. (for 0·1 per cent proof stress on 2 in.), the particular load will give the proof stress.

For example, Table V shows that the extension between 0·28 and 0·30 tons is 4 dial divisions, and between 0·30 and 0·32 tons, 6 divisions. The amount of permanent extension between these two loads is therefore $4 - 3 = 1$, and $6 - 3 = 3$ divisions respectively. If these calculations be made from the load increment at which an extension in excess of the 3 divisions elastic occurs and the amount added until a total as near to 40 as possible is reached, then the proof stress will be known to close limits.

The separate amounts of permanent extension are given in Table V.

TABLE V. LOAD/EXTENSION TEST RESULTS

Load	Extension*	Total extn. per 0·02 ton	Permanent extn. per 0·02 ton	Total permanent
tons	dial divs.*	divs.	divs.	divs.
·10	12½	—	—	—
·12	15	2½	—	—
·14	18	3	—	—
·16	21	3	—	—
·18	24	3	—	—
·20	27	3	—	—
·22	30	3	—	—
·24	33½	3½	½	½
·26	37	3½	½	1
·28	41	4	1	2
·30	45	4	1	3
·32	51	6	3	6
·34	58	7	4	10
·36	70	12	9	19
·37	81			
·38	95	25	22	41
·39	110			
·40	130	35	32	
·41	150			

* Extension on 2 in. gauge length.
1 dial division equals 1/20,000 in.

From the table it is evident that the proof stress is very close to 0·38 tons load and the actual plotted graph gave a value of 0·379 tons. This value divided by the cross-sectional area of the gauge length will give the proof stress.

When load increments are larger it may be necessary to adopt a slightly different procedure in order to make the approximation as close as possible.

For example, in Table II (page 11), the extension between 11 and 12 tons load is 17 dial divisions, of which 13 represent the elastic part and the remaining 4 the permanent extension. From 12 to 13 tons load the extension is 61, and 61 — 13 will again represent the permanent part, i.e. 48 divisions.

The 48 added to the previous 4 gives 52 divisions as the total amount of permanent extension at 13 tons load. On the particular dial in question 40 divisions equal 0·002 in., so it is evident that the 0·1 per cent proof stress has been exceeded at that load.

A closer approximation will be given by referring to the extension from 12 to 12·9 tons load, i.e. 39 divisions. If 13 divisions represent the extension for 1 ton load, then for 0·9 ton, $0{\cdot}9 \times 13 = 11{\cdot}7$, say 12 divisions will be given. Therefore $39 - 12 = 27$ divisions will be the amount of permanent extension between 12 and 12·9 tons load. Add to this the 4 divisions representing the permanent extension between 11 and 12 tons, and the result is 31 divisions.

If, then, at 12·9 tons load there is a permanent extension of 31 divisions and 52 divisions, at 13 tons it is evident that the 0·002 in. required, represented by 40 divisions on the dial, must be obtained between 12·9 and 13 tons load. Reference to the graph of Fig. 7 (page 16) will show that the proof stress is given by a load of 12·95 tons.

Although the above explanation may seem tedious, yet it is a very simple matter to scrutinize test results for the estimation of proof stress. It is immaterial what load increments are used; having ascertained the elastic extension due to equal increments, total up the separate amounts of permanent extension until the nearest figure to that permissible is obtained.

The method is also an invaluable check of the proof stress given by a graph. For the test results of Table II (page 11) it is obvious that the proof stress must be given by a load between 12·9 and 13 tons, but a slight error in plotting, or drawing XY, will give a value outside these limits.

If test results are tabulated before the actual plotting, a moment's scrutiny will indicate the approximate proof stress, denoting whether the particular material is satisfactory or otherwise, long before the graphical result can be obtained.

LOAD/EXTENSION TEST RESULTS

The following test results are given for practice plotting, proof stress determination and calculation of the elastic modulus. It will be found most convenient to use graph paper ruled in one inch, half-inch, and one-tenth inch squares.

TABLE VI. ALUMINIUM ALLOY L 40
0·564 in. dia., 2 in. gauge length

Load	Extension	Load	Extension
tons	dial divs.	tons	dial divs.
0·75	dial to 0	5·10	171
1·50	27*	5·20	179
2·00	45	5·30	189
2·50	64	5·40	199
3·00	82	5·44*	202
3·50	100	5·50	209
4·00	118	5·60	222
4·40	133	5·66*	229
4·60	140	5·70	235
4·80	150	5·80	254
5·00	162		

1 dial division = 1/20,000 in.
* Readings for four-point determination at limits of 21, 24 tons/sq. in.
SCALE; 1 in. = 1 ton, 1 in. = 40/20,000 in.

TABLE VII. ALUMINIUM ALLOY L3
Test piece ·039 in. × 0·750 in., 2 in. gauge length

Load	Extension	Load	Extension
tons	dial divs.	tons	dial divs.
0·0877	dial to 0	0·38	93
0·175	26	0·40	103
0·263	52	0·42	116
0·300	63	0·44	136
0·325	72	0·45	148
0·340	78	0·46	164
0·360	85	0·47	184

Scale as for Table VI. or load scale 2 in. = 1 ton

TABLE VIII. BRIGHT DRAWN M.S. (COMMERCIAL)
0·564 in. dia., 2 in. gauge length

Load	Extension	Load	Extension
tons	dial divs.	tons	dial divs.
1·0	11½	6·0	84
2·0	24	6·2	90
3·0	36	6·4	98
4·0	48	6·6	107
4·5	54	6·8	119
5·0	62	7·0	130
5·5	71	7·2	147

Scale as for Table VI

TABLE IX. NICKEL CHROMIUM STEEL S.11
0·564 in. dia., 2 in. gauge length

Load	Extension	Load	Extension
tons	dial divs.	tons	dial divs.
2·0	25	10·0	127
3·0	37	10·4	133
4·0	50	10·8	138½
5·0	62½	11·2	145
6·0	75	11·6	154
7·0	88	11·8	158
8·0	100	12·0	164
9·0	113	12·2	172
9·5	120	12·4	185
		12·6	204

Scale 1 in. = 2 tons, 1 in. = 40/20,000 in.

TABLE X. BRASS
0·564 in. dia., 2 in. gauge length

Load	Extension	Load	Extension
tons	dial divs.	tons	dial divs.
0·5	13	3·0	90
1·0	27	3·1	96
1·5	40	3·2	102½
2·0	53	3·3	110
2·5	68	3·4	118
2·7	76	3·5	126
2·9	85	3·6	134

Scale 1 in. = 0·5 ton, 1 in. = 40/20,000 in.

CHAPTER III

TYPES OF TENSILE-TESTING MACHINES AND INSTRUMENTS

THE three main types of tensile testing machines are (1) Single-lever. (2) Multiple-lever. (3) Dial indicating.

The load indications of the single and multiple-lever machines are ascertained from a steelyard in conjunction with a counterpoise in a similar manner to the steelyard type weighing machine, while the third type has a lever system terminating in a dial, from the scale of which the load is self-indicated by a pointer traversing the graduations.

Although all three types are still made, the single-lever machine has been superseded to a great extent by the multiple-lever type, and this in its turn is being superseded by the more recent self-indicating dial type.

The trend of design has been to simplify the operation of testing machines without sacrificing the inherent accuracy of the simple lever steelyard type. The need for accurate proof stress determinations on aircraft metallic materials has played a great part in the design of the modern machine, as for this purpose very small increments of load must be applied, and must be held stable when extensions of the order of 1/20,000 in. are being observed.

SINGLE-LEVER MACHINES

In this machine, the load on the specimen is measured by a simple lever, as in a steelyard weighing machine. The load is applied to one end of the specimen by means of screw gearing or hydraulic power, and is transmitted by the specimen to the weighing apparatus. In order that the weighing lever may be sensitive to small variations in load, its fulcrum consists of a hardened steel knife-edge. The load is transferred from the weighing crosshead to the lever by means of a knife-edge bearing also.

A diagrammatic arrangement is shown in Fig. 16. The lever A with the poise weight B can be balanced about the fulcrum F. Imagine that the lever is in balance, the distance X between knife-edges K_1 and K_2 is 2 in., and that the poise weighs 10 cwt.

Now, if a pull P of 1 ton be exerted on the load knife-edge K_2, a turning moment of 1×2 ton in. will be exerted about the fulcrum of the lever. In order to maintain balance, the poise B must be moved so as to impose an equal and opposite moment. Then, taking moments about the lever fulcrum—

$$1 \times 2 \text{ ton in.} = \tfrac{1}{2}\,(10 \text{ cwt.}) \times D$$

where D = distance poise must be moved.

$$\therefore \quad D = 4 \text{ in.}$$

Therefore the poise weight of 10 cwt. must be moved 4 in. along the lever in order to balance a load of 1 ton applied to the knife-edge K_2.

With such a machine the main scale would be graduated 4 in. = 1 ton. For accurate reading every 4 in. would be divided into 10 equal parts, giving each part a value of 0·1 ton. By means of vernier scale carried by the poise, readings to 0·01 ton could then be made.

In order to limit the movement of the beam during a test, a column carrying two stops is erected at the end of the lever. To secure constant leverage with varying positions of the lever between the two stops, the points of application of the loads and the centre of gravity of the poise must all be on the same straight line.

It will be noted that the closer the two knife-edges are to each other, the higher becomes the mechanical advantage of the lever. Constructional requirements limit the actual distance that can be obtained.

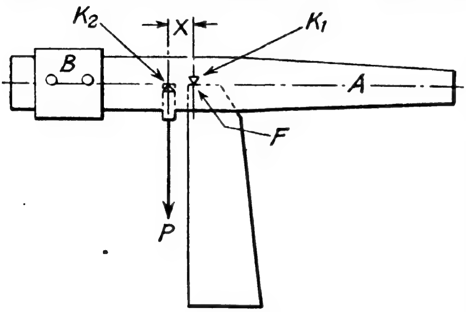


FIG. 16. DIAGRAMMATIC SKETCH OF A SINGLE-LEVER TESTING MACHINE

Some machines have what is termed a "split" poise. This means that a portion of the total poise may be detached and used separately. The advantage of this is that, when testing material of low strength or of small cross-sectional area, the use of the smaller poise enables more accurate readings to be obtained. In other words, the same length of scale, which would represent say 20 tons for the complete poise, would be equivalent to only 4 tons with a poise equal to one-fifth the weight of the complete poise assembly.

Single-lever type machines are normally available in 5, 10, 15, 20, 30, 50, and 100 ton capacities, although smaller and larger machines are available for special requirements. The capacity of the machine is the maximum load which can be exerted and measured.

The Avery-Buckton Single-lever, Type 1015. This machine illustrated in Fig. 17 is described as an example of the single-lever type testing machine. The steelyard is fitted with hardened steel knife-edges, the fulcrum knife-edge resting upon a hardened steel bearing arranged on the standard. The weighing crosshead *C* is suspended from the load knife-edge by means of the link *L*. A single poise *P* is fitted to the standard machine, but this may be made in two parts as illustrated if required. When this is so, the proportion of the combined poise to the small poise is usually in the ratio of 5 : 1 for machines up to 25 tons capacity. The poise is moved along the steelyard by means of the handwheel *H* through bevel and spur gearing arranged at the back of the machine, the layshaft being provided with a universal joint opposite the fulcrum knife-edge to allow free flotation of the steelyard. A balance indicator *B* enables the operator to keep the movement of the steelyard under notice. Coil springs *A* fitted to the end support absorb the shock when the test specimen fractures.

The load on the specimen is indicated upon a machine-engraved scale plate *S* by a vernier *V* attached to the poise, two sets of markings being

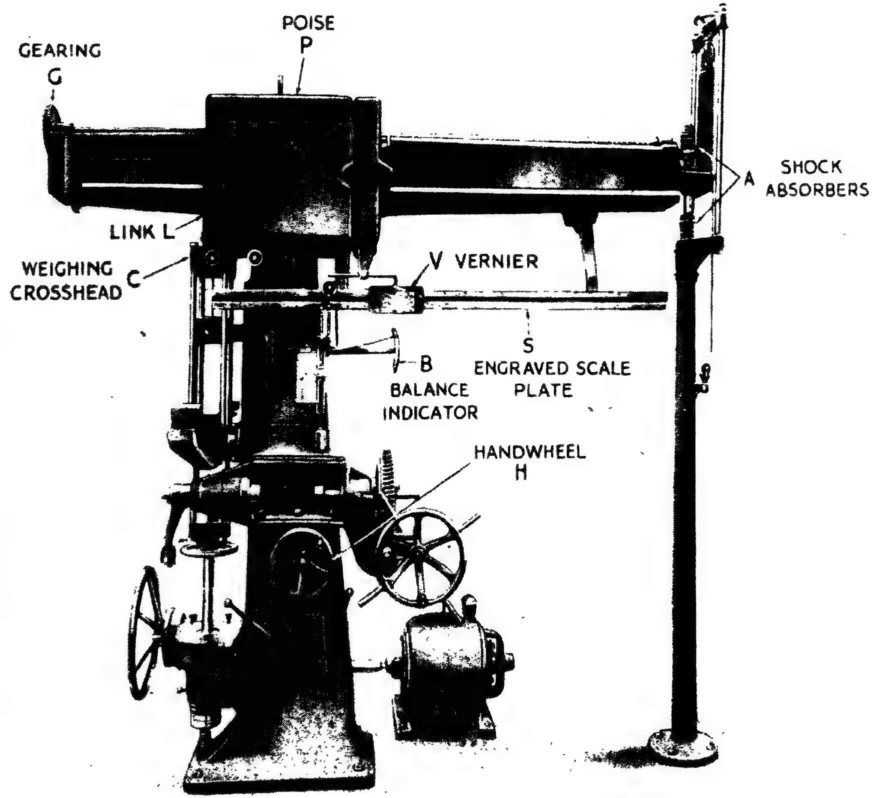


FIG. 17. AVERY-BUCKTON SINGLE-LEVER MACHINE

provided when a split poise is used. One scale, graduated to the full capacity of the machine, is used for the combined poise weight, the other scale being for use with the small poise.

If torsion testing is to be carried out, a third scale graduated in pound-inches is provided.

Straining Mechanism. The straining crosshead, which travels on machined guides, is connected by steel rods to a lower crosshead, to which the straining screw is secured. This steel screw engages with a rotating nut, power-driven by electric motor.

With d.c. supply a variable-speed motor is available, giving straining speeds of $\frac{1}{4}$ in. to 1 in. per minute. For a.c. supplies a constant-speed motor gives a straining speed of $\frac{1}{4}$ in. per minute, and for additional speeds a motor can be provided with two or three fixed speeds. To give a wider range still, two-speed gearboxes in conjunction with a variable-speed motor can be fitted.

For quick adjustment in setting up, the straining screw is provided with a handwheel, and a larger handwheel, driving through worm gearing to the straining screw, enables tests to be made by hand power up to 5 tons load. This hand drive enables very fine increments of load to be added when using an extensometer.

Tests Other than Tensile. With little additional equipment tensile-testing machines can be used for a variety of other tests such as compression, transverse, torsion, shear, and hardness. The machines which can carry out these in addition to the normal tensile test are termed *universal testing machines.*

The fitting for torsion test is bolted to the main standard of the machine and can be seen immediately above handwheel *H* in Fig. 17.

Method of Conducting Tests. Fig. 18 shows a machine arranged for tension, compression, and transverse tests. In any test other than torsion the load is applied to the specimens by the downward movement of the straining crosshead *A*, the load being transmitted through the specimen to the weighing steelyard, where it is balanced off by the poise.

The tensile test piece is positioned between the weighing crosshead *B* and the straining crosshead *A*. The compression specimen is placed on the beam *C*, which is suspended from the weighing crosshead by means of four steel rods.

The load on the test piece, produced by the downward movement of the straining crosshead, is thus transmitted direct to the steelyard, where it is balanced off by the poise as in a tensile test.

Transverse tests can be carried out by supporting the specimen on brackets on the beam *C* and applying a load by means of a presser foot attached to the under side of the straining crosshead.

MULTIPLE-LEVER MACHINES

For some requirements the single-lever machine has certain disadvantages. As the capacity increases, so do the length and weight of the lever, and massiveness of supporting parts. Therefore a comparatively large floor area and headroom are required. Also the inertia of the lever may introduce errors into results unless careful control is exercised. Another disadvantage, particularly on the larger capacity machines, is the

distance through which the end of the steelyard has to drop when fracture of a non-ductile specimen occurs. When fracture occurs with the steelyard in balance the consequent sudden drop and inertia of a large single-lever

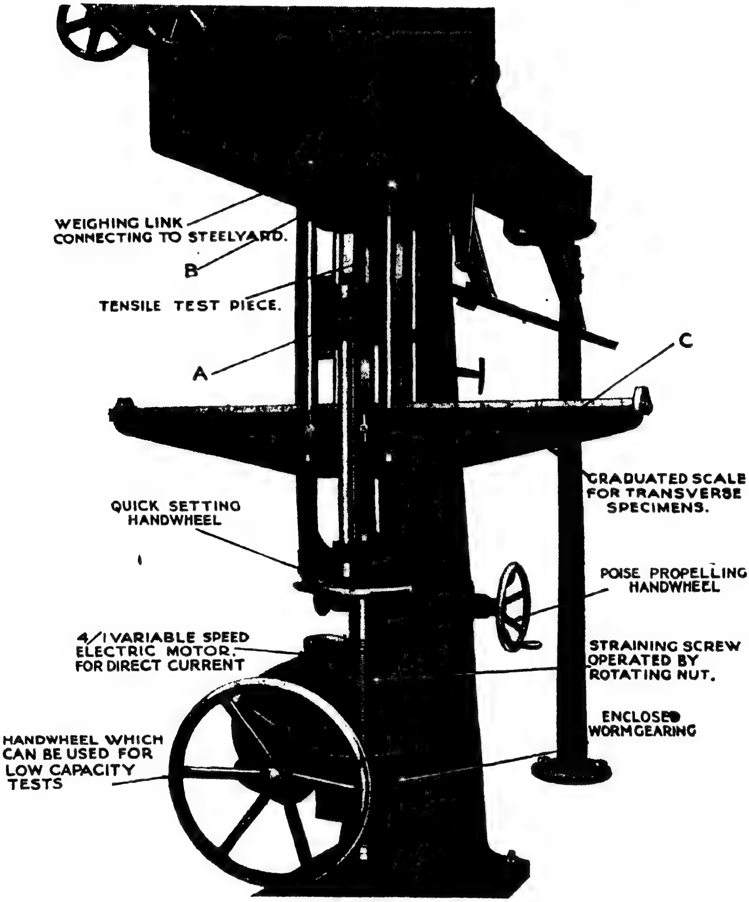


Fig. 18. Avery-Buckton Single-lever Machine (End View)

will cause it to rebound from the shock absorber springs, and apart from the wear and tear to the fulcrum knife-edge, the anticipation of the resulting noise is disturbing to the operator. In these circumstances it is however advisable to limit the travel of the lever. In view of these considerations a machine having a multi-lever system was developed.

In this machine a big mechanical advantage is obtained by using shorter levers linked to each other so as to multiply the leverage. The

complete machine is very much more compact than the single-lever type, and for any given capacity requires considerably less area. In addition, the inertia of the final weighing lever or steelyard is greatly reduced.

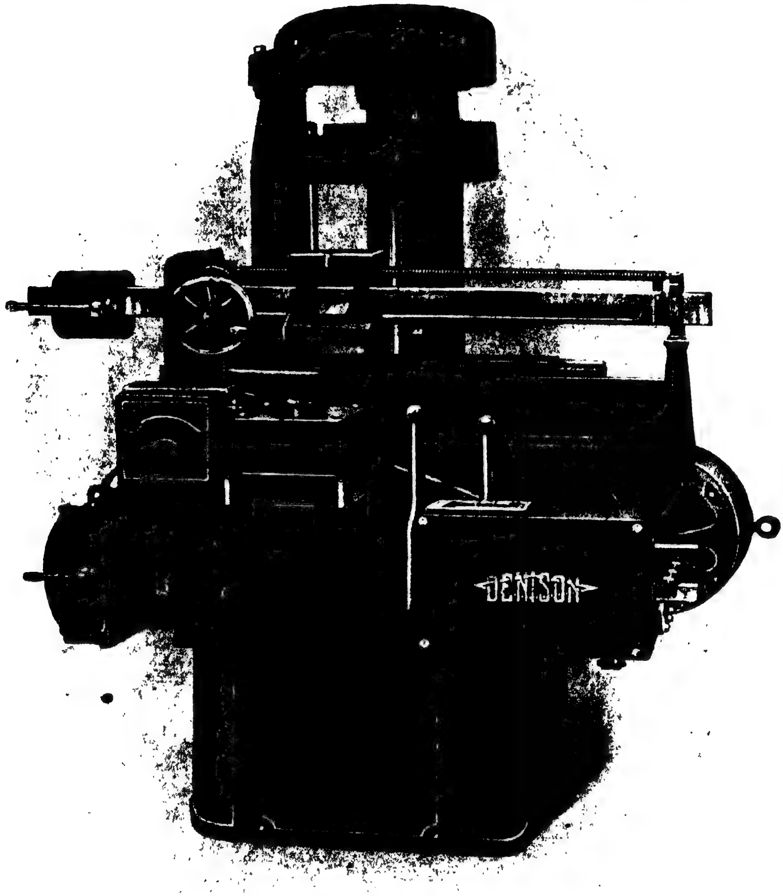


FIG. 19. DENISON ELECTRICALLY-OPERATED MULTIPLE-LEVER MACHINE FOR UNIVERSAL TESTS

The machine can also be easily adapted for long specimens which could not be accommodated in a single-lever machine on account of the height of the standard.

These advantages are gained at the expense of introducing a greater number of knife-edges into the system, all of which must be maintained in first-class condition, but when this is so the multi-lever machine is much

more sensitive and quickly responsive to variations in load than the single-lever type due to the very much less inertia of its parts.

Multiple-lever machines may be either mechanically or hydraulically operated, the particular Denison machine illustrated in Fig. 19 being of the former type. The upper tension crosshead is supported on three columns secured to the weighing table, the straining crosshead also having three columns for stability. There is a single main driving screw driven by a reversible variable speed d.c. electric motor through a four-speed silent worm gearbox, the straining gear being located in the base of the

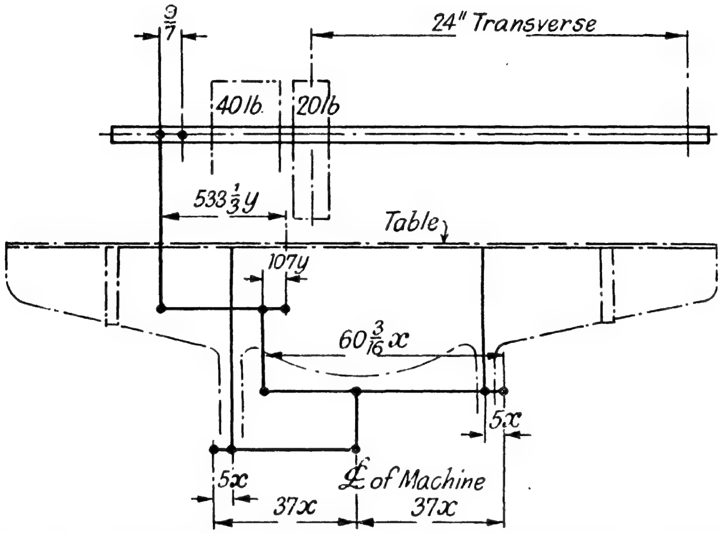


FIG. 19A. DIAGRAMMATIC ARRANGEMENT OF WEIGHING LEVERS—DENISON MULTIPLE-LEVER MACHINE

machine which it is necessary to accommodate in a small pit in the foundations. Alternatively the straining gear can be arranged on the top of the machine, so dispensing with the need of a pit. For the 5, 10, and 15 ton machines of the type illustrated, it is possible to dispense with the pit by slightly reducing the travel of the straining crosshead.

The tensile specimen is located in grips between the upper and straining crossheads and the load applied by downward movement of the latter. The load is then transferred via the test specimen and columns supporting the upper holder to the weighing table which is coupled to the steelyard via intermediate knife-edged bearing levers. A diagrammatic arrangement of the lever system is illustrated in Fig. 19A.

Self-indicating Testing Machines. On these machines the steelyard of the single and multiple-lever type is discarded, the load being automatically indicated on a scale without the necessity for maintaining a steelyard in a "floating" position. This is a great asset when carrying out proof

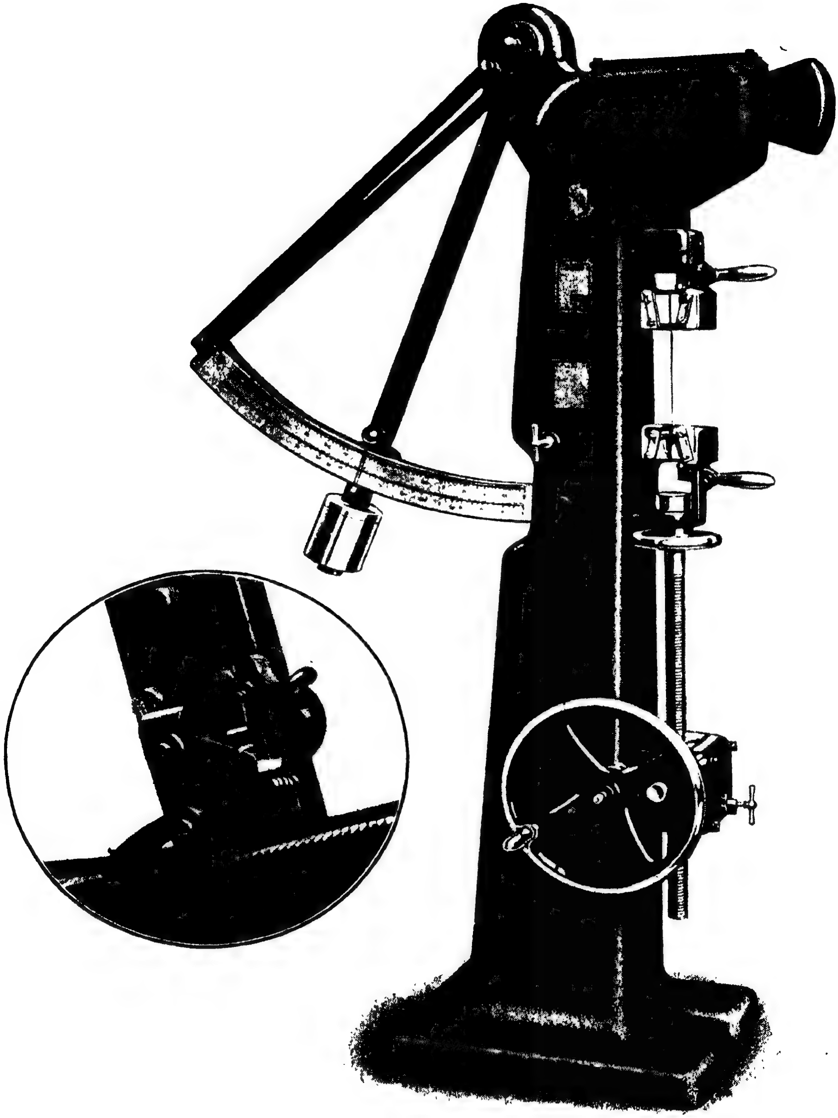


FIG. 20. PENDULUM-TYPE SELF-INDICATING TENSILE MACHINE
(*Avery*)

stress determinations as the operator is relieved of the delicate control necessitated to maintain balance in addition to application of the load.

PENDULUM-TYPE SELF-INDICATING TENSILE MACHINE

The Avery No. 1010 tensile-testing machine illustrated in Fig. 20 is designed for carrying out tensile tests on wire and, by using special grips, proof stress determinations on sheet and strip material.

The load on the specimen is automatically indicated by a pendulum on a quadrant. Two types of straining gear are available, either electric or hand-driven. The hand-driven type is suitable for proof stress

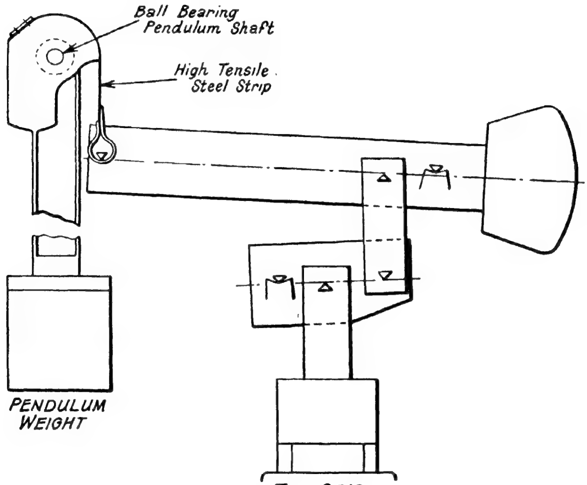


FIG. 21. ARRANGEMENT OF LEVERS FOR PENDULUM MACHINE

determinations, and consists of a hand drive through worm gearing to the straining screw which operates the bottom holder. Quick adjustment of the holder for setting up a specimen is effected by a small handwheel arranged on the straining screw.

Weighing Gear. This is of the pendulum type, the load being transmitted from the top holder to the pendulum by means of two weighing levers fitted with hardened steel knife-edges. The load is finally transmitted to the ball-bearing mounted pendulum by means of a high-tensile steel strip. The arrangement of levers is shown in Fig. 21. The pendulum rises automatically on an accurately graduated scale to indicate the load applied.

Two sets of graduations are provided, the major being from zero to maximum capacity, and the minor from zero to one-quarter maximum capacity. The major and minor scales are of equal length, the latter therefore giving greatly increased sensitivity. This is effected by removing part of the pendulum weight.

Pawls provided on the pendulum engage with a toothed quadrant, thus preventing any backward slip and enabling the pendulum to indicate the maximum load. For the purpose of proof stress determinations, it is necessary to operate the machine with a floating pendulum over

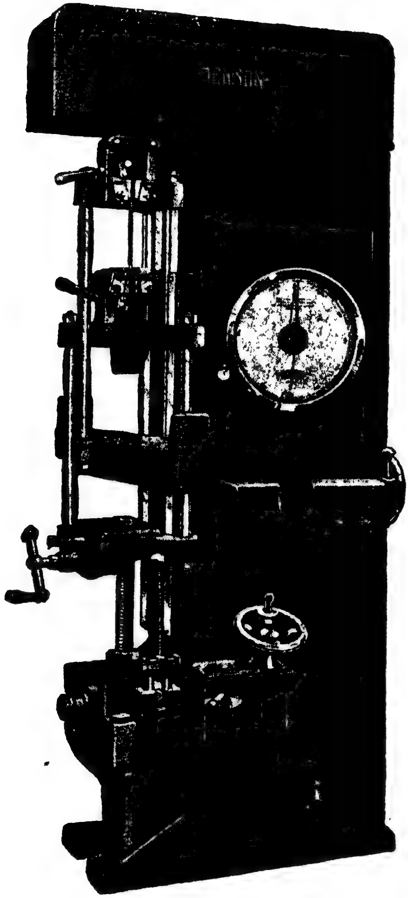


FIG. 22. DENISON TYPE T42 SELF-INDICATING UNIVERSAL MACHINE

that part of the scale used when taking extensometer readings. For this, a catch fitted to the pendulum is provided, which lifts and locks the pawls clear of the rack. With a floating pendulum, both the machine and the operator are more sensitive to variations in load.

Grip Holders. These are of the open quick-operating type, and are supplied complete with three sets of grips. Special provision is made for

controlling the grip by a simple lever action. To prevent misalinement of the pull on the specimen the top holder is balanced and mounted on a knife-edge.

Wedge Grips for Sheet and Strip Metal. For tests on aircraft sheet and strip material a central locating pin in the grips is provided. This locates the test piece on the line of pull by the location holes drilled in the specimen, about which the latter is symmetrical.

Denison Type T42 Self-indicating Machine. This machine, illustrated in Fig. 22, is the smallest of a range of modern testing machines manufactured by Denison, all of which are of the dial-indicating type. One of the special features of this range is the multi capacity, each machine having four capacities, any one of which may be in commission at one time so that in effect there are four machines in one unit. The change over from one capacity to another is effected by the partial rotation of a handwheel at the right-hand side of the column, the ranges being full, one-half, one-fifth, and one-tenth capacity. For proof stress determinations it is therefore possible to use the capacity range which gives the most open scale having regard to the probable maximum load on the test piece, and this is a very desirable feature when carrying out such determinations on thin metal sheet and strip material of varying gauges (up to 10G) when the increments of load are necessarily small and accurate reading of the dial indications is essential.

Fig. 23 shows diagrammatically the arrangement of the weighing levers and cam gear for changing capacity from which it will be noted that the capacity change is effected by alteration of the fulcrum of the lever through which the pull is transferred to the dial mechanism. The particular knife-edge bearing to be used is brought into contact with the lever knife-edge by the action of a cam caused by partial rotation of the handwheel.

By this means the pull on the rod linking the capacity change lever to the indicating mechanism is kept constant for full travel of the pointer. In Fig. 23, with the exception of the capacity change lever, the actual knife-edge bearings are, for the sake of simplicity, illustrated as dots.

The dashpot of the dial unit is for the purpose of preventing shock in this unit when the specimen under test fractures. When the dial pointer has been moved round to any appreciable load there is considerable energy stored in the weighing springs of the dial unit, and these springs would return to, and go beyond their zero position, with great rapidity when the specimen fractured, with consequent undue wear and tear, if they were not restricted.

The piston of the dashpot is spherical and has a clearance of a few thousandths of an inch in its cylinder. Being spherical, there is always a film of oil between the piston and the cylinder, and friction is therefore constant and is of a negligible amount.

Two springs are used, one each side of the lever fulcrum, so that there shall be no possibility of the fulcrum ever jumping from its seat. Further, the use of two springs gives an action comparable in some respects with dashpotting since on the return, as one spring shortens the other lengthens, i.e. its resistance increases.

The dial of the machine has a single circular chart, and windows are

provided in the blanking plate in line with the main graduations, through which are viewed the loads relating to the particular capacity in operation. At the end of the scale an additional window indicates the value of the smallest graduation.

When the capacity change handwheel is operated, the figures in the windows are also automatically changed to denote the loads applicable to the new capacity.

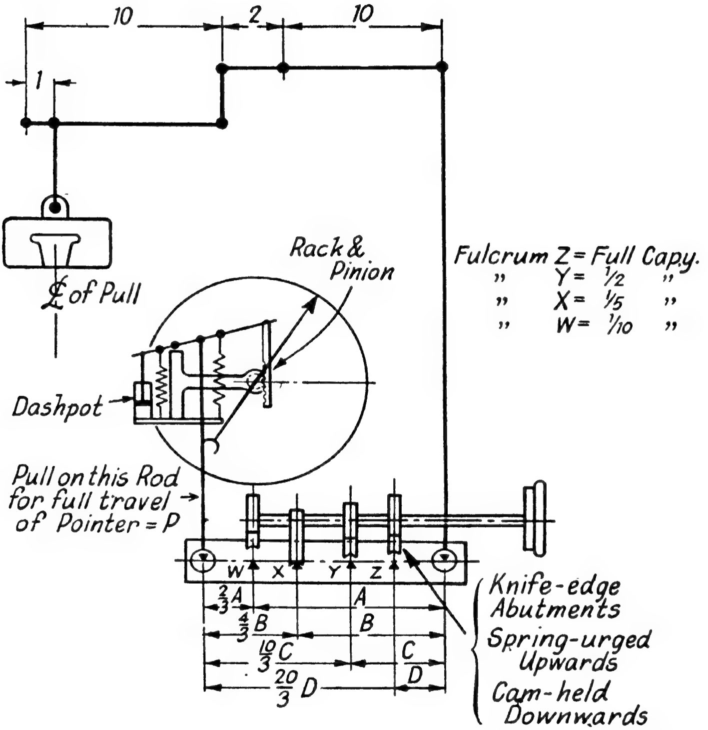


FIG. 23. DIAGRAMMATIC ARRANGEMENT OF WEIGHING LEVERS AND DIAL UNIT—DENISON TYPE T42 MACHINE

The machine illustrated in Fig. 22 is of 5·5 tons maximum capacity, but the type of dial fitted to all machines has the same characteristics and is illustrated in Fig. 24. The long single pointer is loose and is carried around the dial by the main pointer in order to indicate the maximum load.

On the small T42 type machine (Fig. 22) the straining gear is of the mechanical type. An electric motor with inbuilt gearbox drives the main two-speed gearbox of the machine, the gears and base of the straining screw being oil immersed.

For proof stress determinations an auxiliary hand control of the straining gear is provided, and to the left of this is a hand lever which operates

mechanism to clutch either the motor drive or the hand drive to the straining mechanism.

Although the machine illustrated is arranged for universal testing—tensile, compression, transverse, shear, hardness, punching and cupping tests—a model for tensile testing only is available.

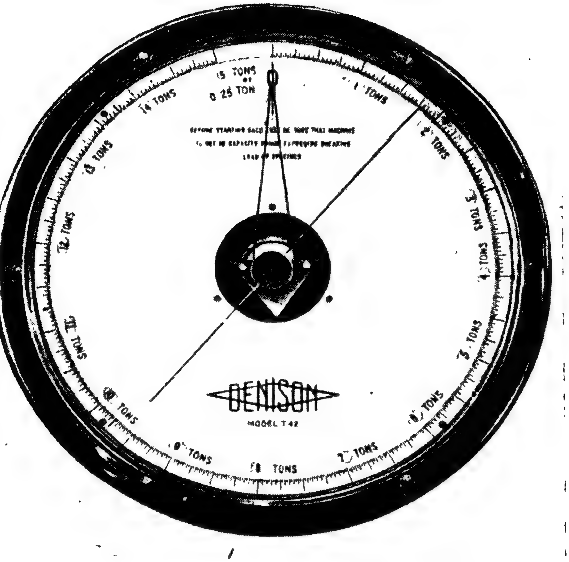


FIG. 24. INDICATING DIAL OF DENISON MACHINES

ALL-HYDRAULIC SELF-INDICATING MACHINES

These machines represent another development in testing equipment, and possess several advantages over the lever type in which a steelyard is used for final indication of the load. In the ordinary hydraulically operated type of lever machine, the hydraulic power is used solely as the means by which the necessary load is obtained. The actual weighing of the load is carried out by transference through the test piece to the steelyard.

If no friction existed between the ram and the straining cylinder, the load could be measured equally well by multiplying the pressure of the fluid in the straining cylinder by the effective area of the ram. This is made possible in the later-type machines by making the ram such a precision fit in the cylinder that no packing is required to maintain pressure-tight operation, as was previously necessary. The very small working clearance provided allows a lubricating film of oil to be maintained, and friction is negligible.

The Method Used to Indicate the Load. It is not possible to obtain the load direct from the very high pressure existing in the straining

cylinder, so that use is made of what is termed a proportional ram. This is in communication with the main straining cylinder, but it has an area considerably smaller than that of the main ram. Therefore, although the same pressure may be exerted on both, the total force transmitted by the smaller ram is much reduced. For example, a pressure of 2000 lb./sq. in. acting on a ram area of 10 sq. in. would be equivalent to a load of 20,000 lb. on the test specimen; but if the area of the proportional ram

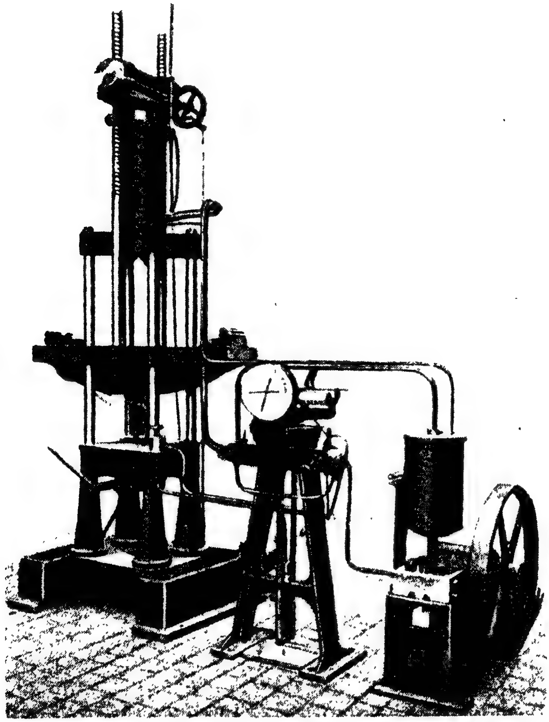


FIG. 25. AMSLER 50-TON HYDRAULIC SELF-INDICATING UNIVERSAL MACHINE
(*T. C. Howden & Co.*)

were only 0·5 sq. in., then the force transmitted by it would be reduced to one-twentieth of 20,000 lb., i.e. 1000 lb., as 0·5 is one twentieth of 10. It is this greatly reduced load which actuates the mechanism connected to the direct-reading dial.

The Amsler Testing Machine. This is shown in Figs. 25 and 26, from which it is seen that the complete installation consists of three units—

1. The actual testing machine.
2. A pendulum dynamometer.
3. High-pressure pump.

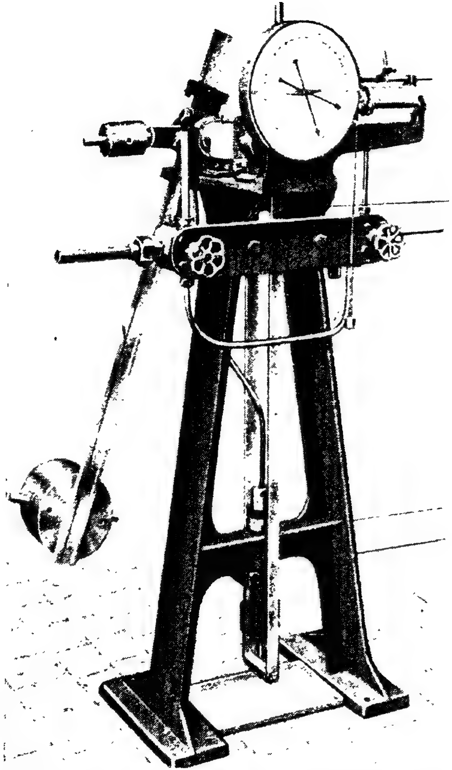


Fig. 26a. Pendulum Dynamometer for Amsler Testing Machine

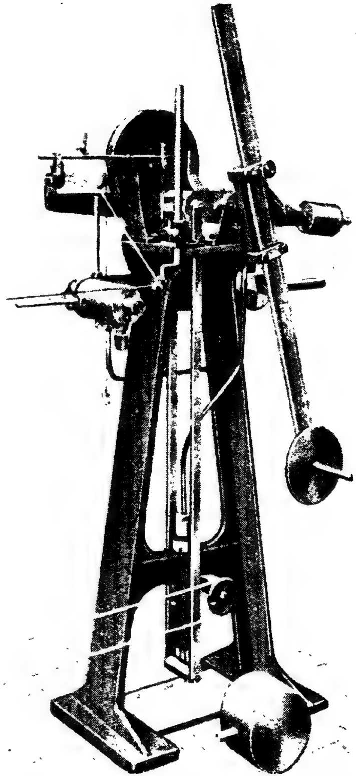


Fig. 26b. Rear View of Fig. 26a

(T. C. Howden & Co.)

The straining cylinder is located at the top of the testing machine, the ram being fitted to the uppermost end. The ram carries a cradle from which the straining crosshead is suspended, the four columns supporting the straining cylinder acting as guides.

The pendulum dynamometer (Fig. 26A) is the load-measuring unit, the load being indicated according to the pressure of oil in the straining cylinder. The proportional ram transmits its thrust to a yoke piece, from which the load is transferred through connecting straps to a lever situated at the top of the frame. This lever is secured to a shaft mounted on ball bearings.

The pull on this lever results in an angular movement of the shaft, which carries at its outer end a clamp. To this clamp the pendulum arm and weight are secured. The pendulum is deflected until it automatically balances the pull on the lever resulting from the action of the oil pressure on the proportional ram. To indicate the load, a cranked arm is secured to the lever, which operates the dial pointer through the medium of a screwed rod and pinion. An extension of this rod also operates the load recorder of the autographic apparatus situated to one side of the load-indicating dial.

In order to eliminate any friction which might exist between the proportional ram and its cylinder, the ram is rotated by means of a pulley, driven either by hand or from the oil pump. To prevent the pendulum from falling back to the vertical position too quickly when the test specimen breaks, a by-pass valve is fitted in the pressure-pipe connection to the top of the proportional ram cylinder.

The sensitivity of the dynamometer can be altered by moving the pendulum shaft in its clamp. Three positions are provided for, with load-indicating dials to suit.

The pressure pump is belt-driven from an electric motor, the oil being supplied from a tank attached to the pump stand. Oil is supplied to the straining cylinder via a valve body on the pendulum dynamometer, and return pipes provide for return oil from under the main ram when the load is released, and for draining any oil leaking past the ram.

The tensile machine is capable of carrying out all the usual tests: compression, bending, shearing, and hardness.

Avery A806/1474. Self-indicating Universal Testing Machine. This machine is of single cylinder construction, the controls being so arranged that the operator can remain seated throughout a test.

Fig. 27 illustrates the 10-ton full-capacity type with the various special features of the machine. Hydraulic loading, hydraulic transmission, and a heavy pendulum load resistant provide the basis of the design.

The four capacity feature of the modern testing machine is incorporated, the ranges being one-half, one-fifth, one-tenth, and full capacity, all ranges being provided on the one chart of the indicator dial. These ratios are retained in the 30-ton and 50-ton machine. The particular chart range in use is denoted by the position of the capacity change control wheel.

Operation of the Machine. A diagrammatic arrangement of the machine is illustrated in Fig. 28, to which reference should be made in order to follow the principle of operation.

The Oil Supply Circuit. High and low pressure pumps are mounted on a common shaft and housed in an oil reservoir at the base of the indi-

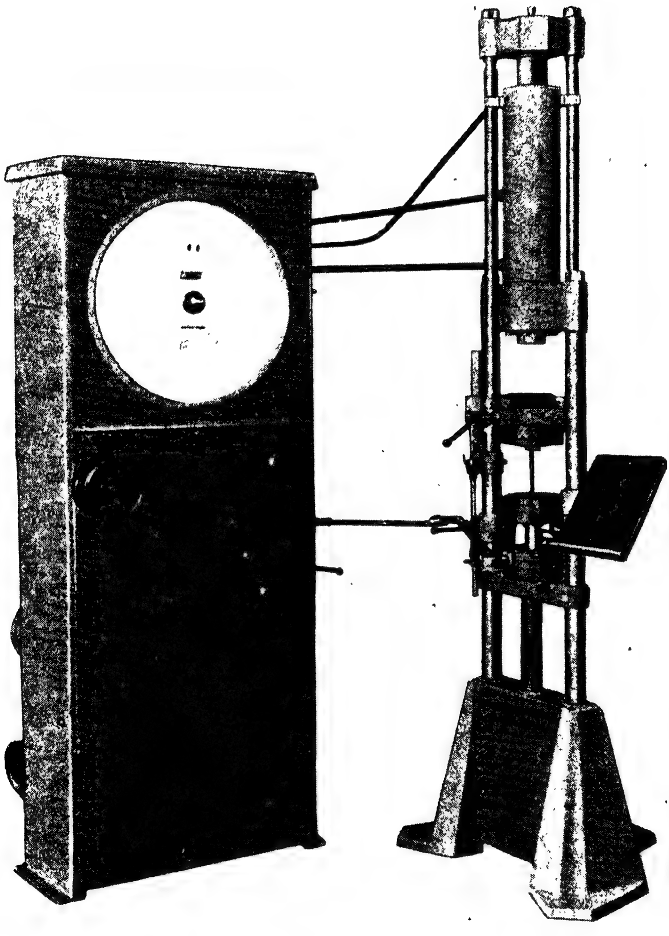


FIG. 27. AVERY TYPE A806/1474 10-TON SELF-INDICATING UNIVERSAL MACHINE

cating unit. An electric motor mounted on the floor behind the cabinet drives the pump unit via a belt drive, operation of the motor being by push button control on the front of the cabinet.

The high pressure pump supplies oil continuously to the main cylinder

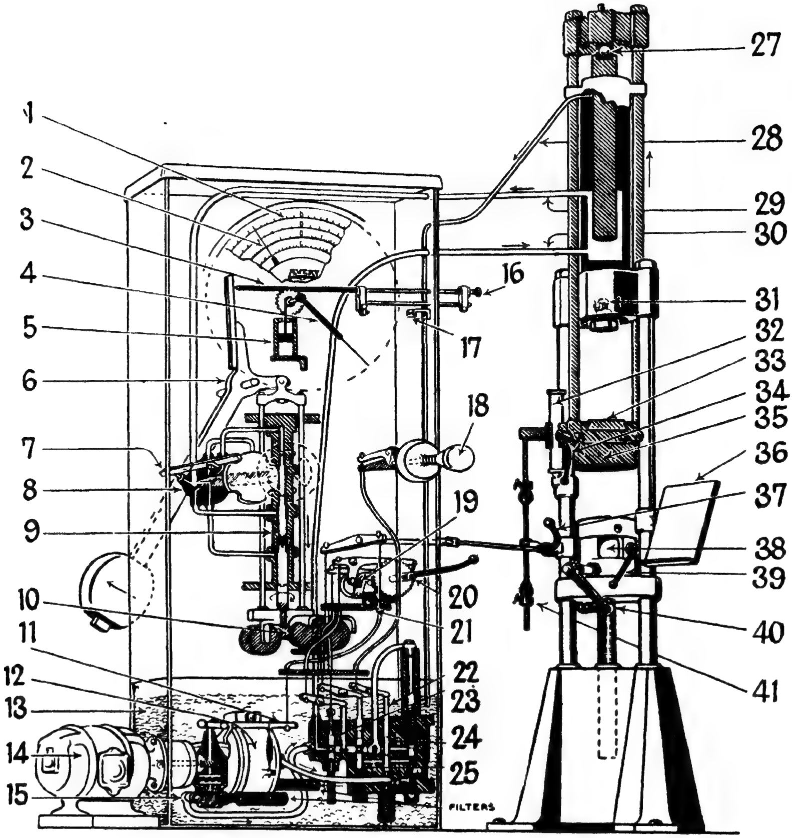


Fig. 28. Diagrammatic Arrangement of Avery Type A806/1474 Machine

1. Quadruple Capacity Chart.
2. Maximum Load Pointer.
3. Indicator Rack and Pinion.
4. Indicating Pointer.
5. Indicator Dashpot.
6. Pendulum Resistant.
7. Pressure Locking Lever.
8. Four Way Valve.
9. Quadruple Proportional Cylinder Unit.
10. Motor Drive for Proportional Pistons.
11. Clutch Lever.
12. High Pressure Pump.
13. Oil Reservoir.
14. Standard Motor, Floor Mounted.
15. Low Pressure Gear Pump.
16. Zero Setting Knob for Balance.
17. Overload Switch.
18. Strain Setting Control.
19. Strain Mechanism Trip.
20. Load Holding Lever.
21. Vernier Adjustment.
22. Control Valves.
23. Release Valve.
24. Taper Piston Valve.
25. Load Holding Valve.
26. Filters.
27. Top Crosshead Ball Seating.
28. Drain Pipe.
29. Connection to Proportional Cylinders.
30. Supply Pipe to Cylinder.
31. Upper Compression Platen Locator.
32. Extension Scale.
33. Lower Compression Platen.
34. Upper Grip Control.
35. Upper Grip Holder.
36. Adjustable Desk for Note Pad.
37. Quick Setting Control Operates Low Pressure Pump Clutch or Release Valve—Two Directions.
38. Lower Grip Holder.
39. Lower Grip Control.
40. Elevating Lever of Lower Grip Holder.
41. Limit Stops for Upper Grip Holder.

of the straining unit via a control valve block also submerged in the oil tank, the supply being controlled by a taper piston valve operated by the strain setting control.

Between the taper piston valve controlling the supply and the main cylinder supply circuit is a balanced stop valve operated by the straining control lever, and no high pressure supply can pass to the machine when this lever is in a vertically downwards position. The lever must be moved through 180° anti-clockwise in order to bring the straining mechanism into operation, in which position it is held by a detent.

The strain setting control dial is used to determine the straining rate, and four complete turns of the dial are required to cover the full straining range. Annular grooves on the dial spindle indicate on which revolution the control is set.

A balanced piston valve in this circuit releases the supply back to the tank by operation of the quick setting control.

Quick Ram Setting. The low pressure gear pump, giving approximately ten times the output of the high pressure pump, is driven through a spiral drive from the main pump shaft, and in normal straining the gears are idle. The quick setting control, with the lever in the highest position, operates a clutch to drive the gear pump which results in oil being supplied through a non-return valve to the main cylinder of the straining unit, at sufficient pressure to overcome the weight of the ram and crossheads.

The lowest position of this lever opens the release valve, hence to raise the ram lift up the lever, and to lower the ram depress the lever (the intermediate position being neutral).

The lever is provided on the left-hand side of the straining unit to enable the operator to control the machine with the left-hand, leaving the right hand free for handling the grips and the test specimen.

Automatic Setting. A trip rod secured to the upper grip holder carries adjustable limit stops. These can be set to position, the grip holder in either upward or downward movement by tripping detents arranged on the quick setting lever which return the lever to its neutral position, i.e. release valve closed and low pressure pump clutch disengaged.

As the resetting of the machine takes place during the removal of the broken specimen this feature is of particular value in repetition testing.

Load Holding. The strain lever clutching in the high pressure oil can be used also for temporarily holding loads by moving the lever to an intermediate position. Coarse adjustment is effected by the lever and the vernier knob below the lever for finer control.

Weighing Circuit. Connection to the proportional cylinders from the main cylinder is via a four-way valve, operated by a handwheel control. This control selects in turn any of four proportional cylinders, the capacity being indicated on the control disc.

The control cannot be operated unless all pressure is removed from the machine owing to the incorporation of a pressure locking lever which prevents change-over being made during the process of a test.

The proportional cylinder unit comprises four cylinders mounted each above the other in the relationship one-tenth, one-fifth, half, and full capacities, i.e. the cylinder cross sectional areas are in these ratios. A

tongue and groove connects each ram end to end and they are driven by a geared motor at 44 r.p.m. to minimize friction.

The rams push on to a yoke which operates a heavy pendulum resistant, the pull on the pendulum being the same for each capacity and is 336 lb. For example, the lowest cylinder is for one-tenth capacity operation, but its cross-sectional area is ten times that of the uppermost or full capacity cylinder. If therefore the pressure in the main straining cylinder is only one-tenth that for full capacity the proportional ram, having ten times the area, will transmit exactly the same force for the full range of the chart. In other words pressure × area has a constant value.

A hardened and lapped steel blade attached to the pendulum pushes the indicator rack and operates the indicator pinion and weighing pointer.

When a change in capacity is made, and the upper grip holder raised hydraulically until floating on oil, the weighing pointer will move off zero. This is due to the initial dead weight effect of the ram and upper crosshead on the different proportional cylinder areas, and the zero is re-set by means of the balance knob shown in Fig. 28.

Chart Capacities. The chart for the 10 tons machine illustrated in Fig. 27, is shown in greater detail in Fig. 29, from which the fineness of graduation will be apparent.

A sensitive maximum load pointer is operated by the weighing pointer and is reset manually by the small knurled knob in the dial centre.

Denison Self-indicating Universal Machines Type T42A, T42B. These machines combine steelyard type weighing with dial indication and hydraulic straining.

The essential feature of the design is that the load on the specimen is truly weighed and is not taken from measurement of the oil pressure in the straining cylinder.

The upper end of the specimen connects to the upper wedgebox which is carried from a system of knife-edged bearing levers terminating in the indicating dial. The lower end of the specimen is connected to the hydraulic straining system via the lower wedgebox.

The hydraulic system is perfectly orthodox, the ram using ordinary packings, this being possible since no measurement is made of the pressure in the cylinder and the variation in friction between the ram and cylinder, ramrod and gland, and between the crosshead guide and the stand, has no effect whatever on the load upon the specimen nor upon the indication of that load.

Each type has a four capacity range and the capacity cam figure changing mechanism described previously in relation to the smaller T42 machine.

Type T42A 10 or 15 tons Maximum Capacity. At the base of the machine (Fig. 30) is the hydraulic cylinder fitted with a piston and piston rod, the latter connecting to the straining crosshead which in turn connects to the straining wedgebox. The cylinder is fed with oil via a distributor valve from a constant speed variable displacement pump, the discharge from which is variable from zero to maximum by infinitesimal increments and decrements by turning the control handwheel.

The distributor valve has four positions, namely (1) Return, (2) Neutral, (3) Constant, (4) Loading and Unloading.

With the valve set to the "loading and unloading" position a small amount of "bleed" is permitted from the loading side of the cylinder

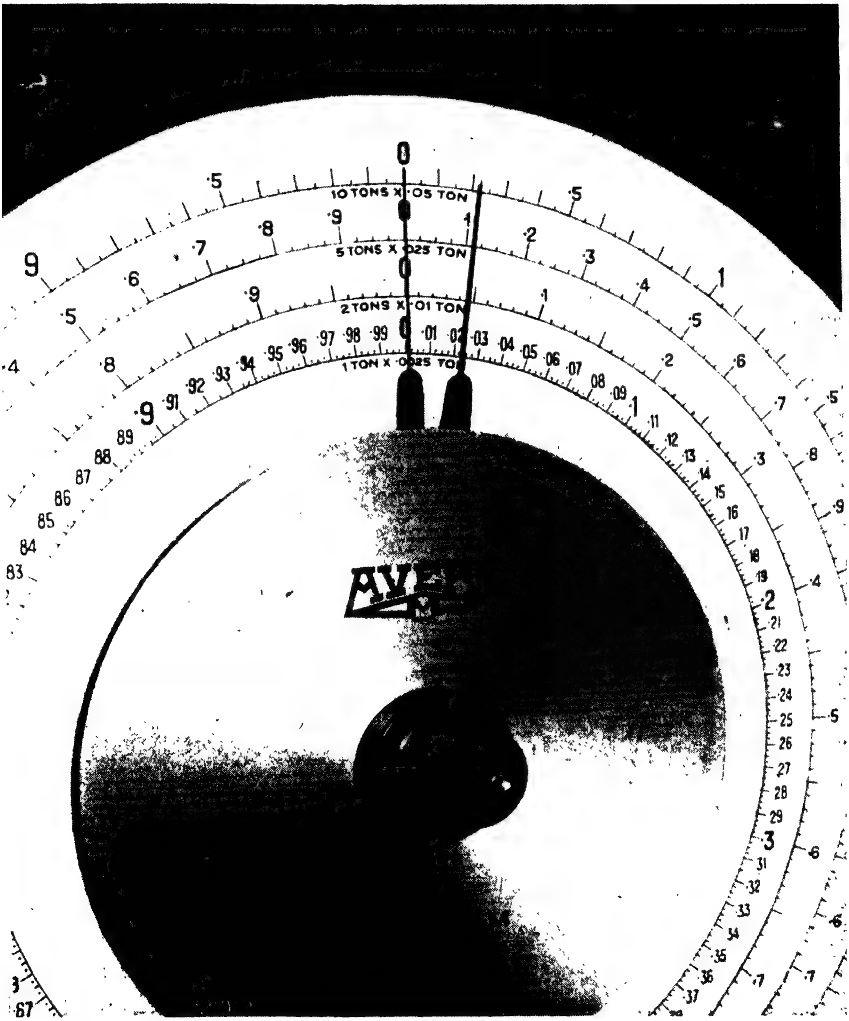


FIG. 29. INDICATING DIAL ON AVERY TYPE A806/1474 MACHINE

Operation of the pump control wheel with the valve in this position permits of loading and unloading the specimen, with stoppages at any point for any length of time, with the load held, provided that the material is not outside the elastic range.

It will be clear that if the bleed is such as to permit X cubic inches per minute of oil to escape from the cylinder when there is a certain load upon the specimen, then if the pump control handwheel is so set that the pump displacement is X cubic inches per minute, the hydraulic

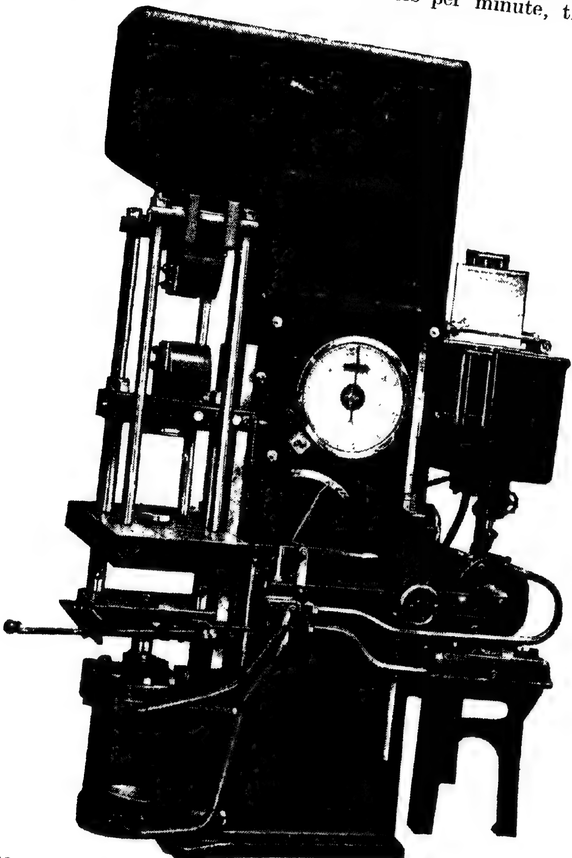


FIG. 30. DENISON TYPE T42A SELF-INDICATING UNIVERSAL MACHINE

piston will neither advance nor return, and the load in the specimen will remain constant. If the pump handwheel is now turned to increase the displacement then the pump discharge to the cylinder exceeds the amount of bleed, the piston moves and the load increases. If the handwheel is turned the opposite way so that the pump is delivering less than X cubic inches per minute, the amount of bleed exceeds the amount fed to the

cylinder with the result that the piston moves upwards and the load on the specimen is reduced.

Type T42B. This machine is of 30 or 50 ton maximum capacity and has weighing and indicating gear exactly similar to the T42. As seen from Fig. 31 the transverse table is arranged in plan, slightly diagonally so that

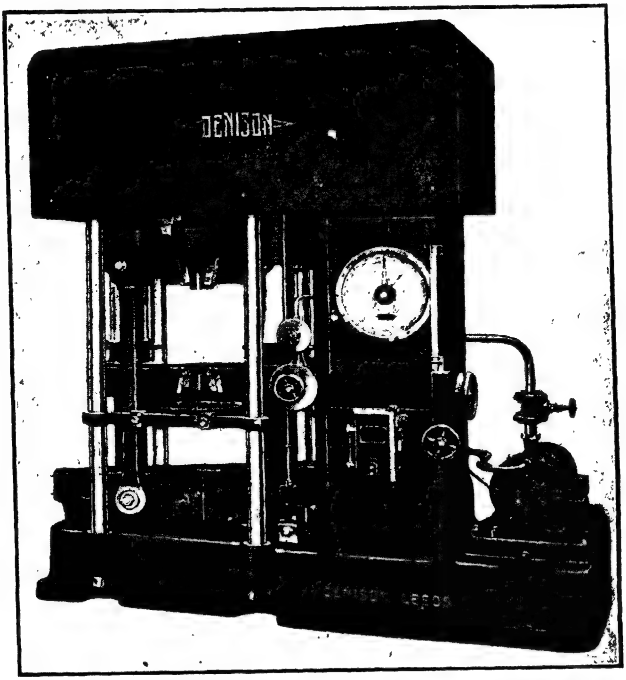


FIG. 31. DENISON 30/50-TON TYPE T42B SELF-INDICATING UNIVERSAL MACHINE

extra long transverse test pieces can be put in the machine, the test piece extension being behind the dial column. If desired an extra girder can be placed upon the table and the transverse tests supported upon this for long spans.

The main weighing lever system shown diagrammatically in Fig. 32 is carried upon two girder members mounted upon four mild steel columns which are in turn mounted upon the base plate of the machine. To these longitudinal girder members are fixed the two straining cylinders.

Two cylinders are used to keep down headroom, and although the machine has a variable distance of from 1 to 24 in., the overall height is only 8 ft. 10 in., no holes being required in the floor.

The rams are single acting and the recovery or return is by means of springs which are totally enclosed and are inside the rams. The constant

speed variable displacement pump is controlled by the small handwheel on the front of the dial column, and further valve control is provided by means of two handwheels on the left-hand side of the front of the dial column (Fig. 31). The lower valve is the main valve, and the upper a fine control valve.

For straining, both control valves are closed and the speed of travel is varied by altering the pump displacement, i.e. by operating the pump

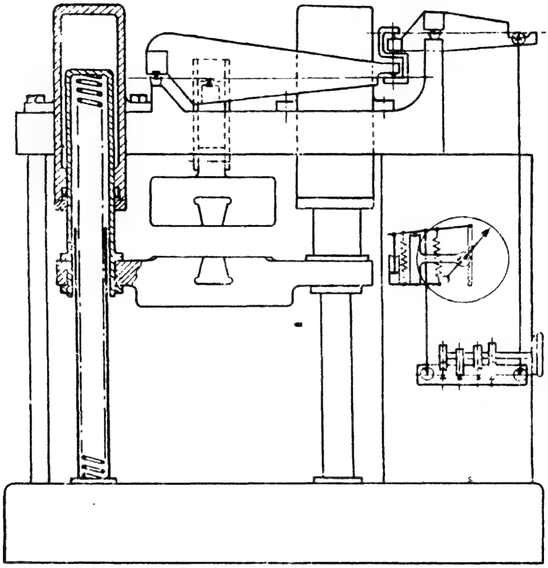


FIG. 32. DIAGRAMMATIC ARRANGEMENT OF WEIGHING LEVER SYSTEM, CYLINDERS, RAMS AND WEDGE BOXES ON DENISON T42B MACHINE

control handwheel. To return, the lower valve is opened, this permitting the springs in the ram to push the rams upwards, the oil passing from the two cylinders through the valve to the tank.

For tests where loading and unloading is desired the lower valve is closed and the upper one opened slightly. This gives a variable amount of bleed, and then either the bleed can be varied keeping the pump displacement constant, or the pump displacement varied keeping the bleed contant. With either method a constant load can be maintained on a specimen, or it can be loaded or unloaded at any desired speed within the maximum provided.

Testing-machine Grips. For all tensile tests it is important to ensure that the line of pull is along the axis of the test specimen. If this is not so, a bending stress will be applied, there will be an unequal distribution of stress across the section, and inaccurate readings will result. In order

to ensure that the load is applied truly axial, provision for self-alignment of the grip holders is usually made.

For plain bar specimens, or standard specimens with plain turned ends, wedge grips are used, the serrated faces of which "bite" into the bar, the wedging action of the complete grip in its holder under load, causing a firmer grip as the load increases.

It is very convenient to have the half grips coupled together so that they can be raised or lowered simultaneously when inserting the specimen

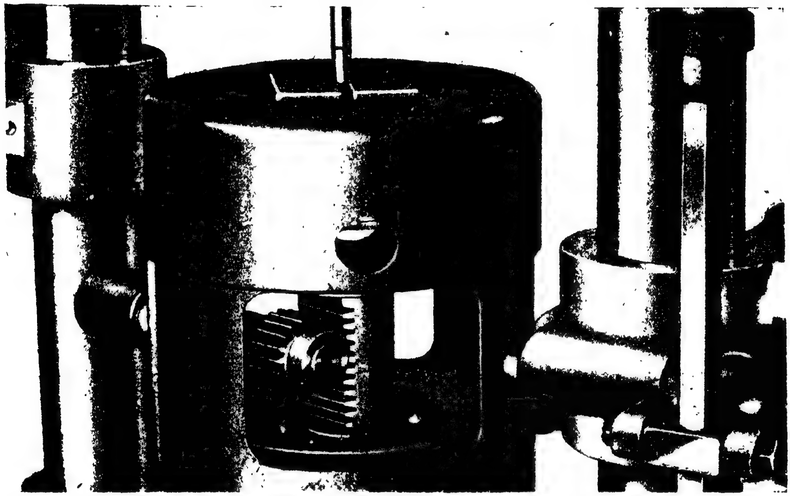


FIG. 33. RACK-OPERATED GRIP HOLDERS ON AVERY A806/1474 MACHINE

and for the purpose of ensuring that they are in alignment with one another. The grip holders (Fig. 33) on the Avery A806/1474 machine are equipped with a lever operated rack to facilitate this operation.

On some machines the wedgeboxes are of the open-fronted type which permits very easy insertion or removal of the test specimen, and in addition the operator can see that the specimen is being held over the complete length of the grip. The Denison T42 type machines are all fitted with the open-fronted type wedgebox, and that fitted on the T42B can be seen in Fig. 31, the wedges being opened and closed by means of a removable key, not shown in the illustration.

The Avery A806/1474 machine also uses this type for strip specimens to BSS485 (page 123 (*a*)), the grips being provided with the necessary $\frac{1}{2}$ in. diameter centralizing peg. The holders are inserted into the machine and aligned by means of a master gauge. A specimen set up for proof stress determination is illustrated in Fig. 34.

Screwed or headed test bars are used when the maximum stress is over about 65 tons/sq. in. as the hardness of the material may prevent the grips from obtaining sufficient purchase to withstand the load. When

proof stress determinations are being made it is also advisable to have screwed or headed bars as any slight movement of a wedge grip whilst extensometer readings are being taken, will upset the accuracy of the reading due to the slight shock which is transmitted to the extensometer.

The supplementary holders for these specimens are accommodated in the main wedgeboxes by removing the wedge grips, and Fig. 35 illustrates types of adaptors used.

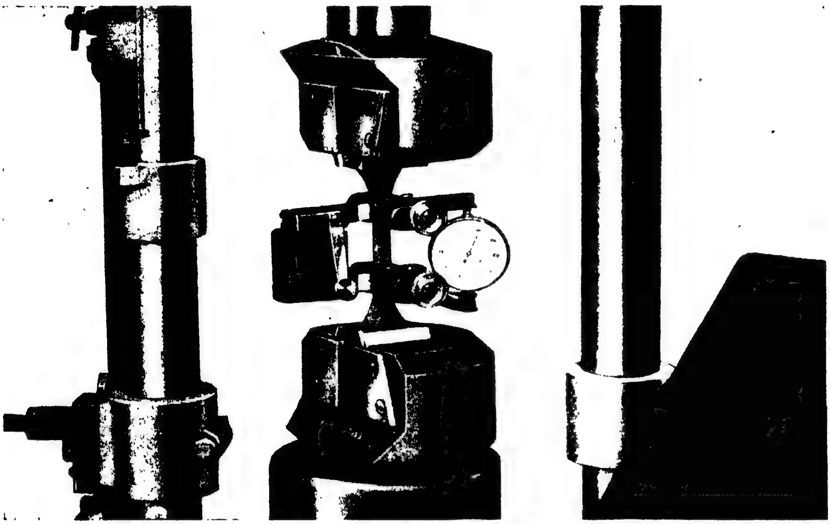


FIG. 34. OPEN FRONTED TYPE WEDGEBOX AND WEDGE GRIPS (*Avery*)

For tensile testing of wire ropes it is usual to prepare conical ends on the wire to facilitate gripping in the special holders provided. The wire is splayed out on the mould shown in Fig. 36, the half-moulds clamped together, and solder run in.

Compression tests can be carried out through the medium of spherically seated platforms which ensure that the load is evenly distributed across the section.

Generally, the particular type of attachments used depend upon the nature and requirements of the individual test. For example, the four-point loading apparatus illustrated in Fig. 37 is the type employed for the determination of the modulus of elasticity and modulus of rupture for timber. This type avoids failure due to the shearing of the fibres which may be caused if central loading on the span is carried out.

While standard forms of testing accessories meet most requirements, it is not a difficult matter to design special types to meet individual requirements if means for self-alinement or definite location are provided.

THE EXTENSOMETER

Mention has already been made of the need for an accurate extension-measuring instrument when determining proof stress, etc. Such an instrument, called an *extensometer*, should measure correctly to 0·005 per cent of the gauge length used, i.e. for a 2 in. gauge length to 0·0001 in., and possess a sensitivity of 0·00005 in.

The construction of an extensometer to fulfil these requirements consistently is no easy matter, as in addition several other factors have to be considered. Weight, robustness, balance, ease of reading, and the arrangement to allow the instrument to be used on different sizes of test pieces, have all to be correlated. It is the combination of all these features in a simple practical instrument that is so difficult, and it is noteworthy that the extensometer subsequently described is the outcome of years of experience in testing requirements.

The shorter the gauge length used, the greater the degree of accuracy. This will be obvious when considering the percentage permanent extension allowable for proof stress determination. A 0·1 per cent proof stress on a 2 in. gauge length is equal to 0·002 in. permanent extension, whereas with a 4 in. length the allowable permanent extension would be 0·004 in. The accuracy of measurement as indicated above must be 0·0001 in. for a 2 in. gauge length and 0·0002 in. for a 4 in. length.

There are optical-type extensometers, suitable for laboratory use only, which will measure extensions of 0·000004 in. (1/250,000 in.).

The Lindley Extensometer. This extensometer, although robust, simple, and practical, is nevertheless capable of accurately measuring extension to ± 0·00005 in. (± 1/20,000 in.), while alterations in length of 0·00001 in. (± 1/100,000) are clearly indicated.

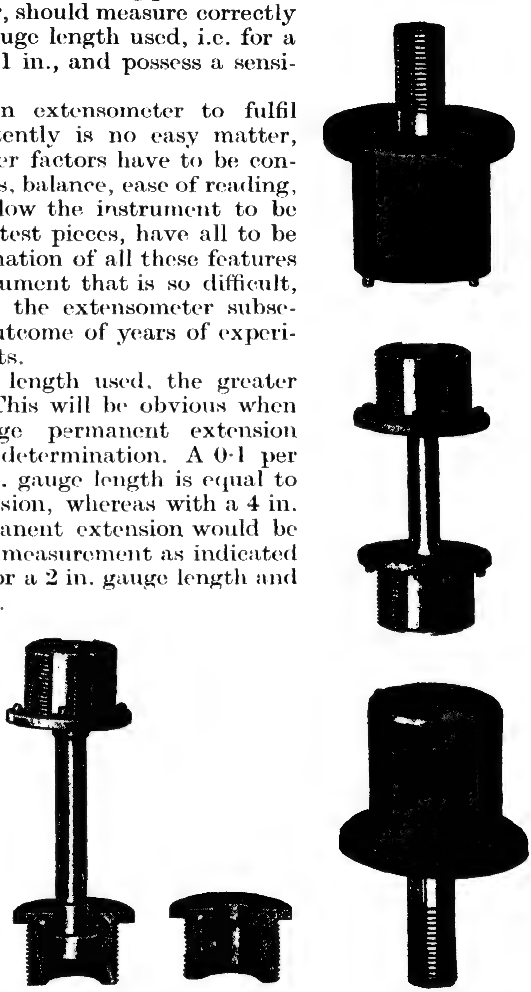


FIG. 35. ADAPTORS FOR HEADED AND SCREWED TEST SPECIMENS
(*Avery*)

The principle on which this extensometer works is simple. The extension of the material under test is magnified in the ratio of 2 : 1 by a single lever, and thereafter measured and indicated by a Mercer dial gauge graduated in ten-thousandths of an inch. A movement of the dial finger over one

division indicates an alteration in the length of the 2 in. gauge length of the specimen of 1/20,000 in. The special features of the Lindley device are—

1. The extensometer is always in static balance about the axis of the test piece, so that no errors are introduced by the weight of the instrument bending the specimen.

2. Minute distortions inevitably set up in the extensometer while gripping and testing do not affect the dial reading.

3. Frictional lag has been almost entirely eliminated.

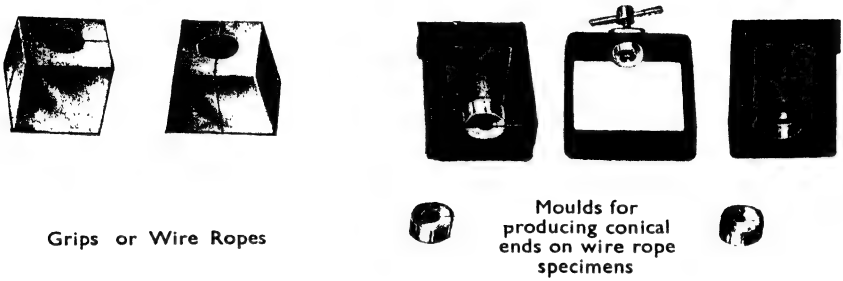


Fig. 36. Moulds for Conical Ends on Wire Rope
(*Avery*)

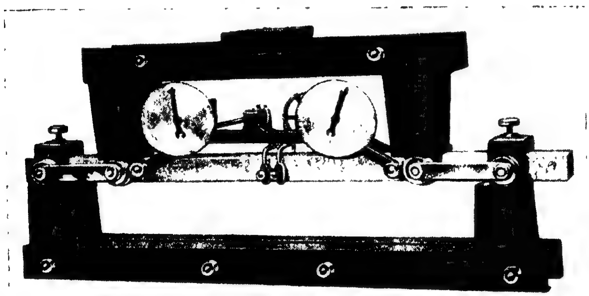


Fig. 37. Four-point Loading Apparatus
(*Avery*)

4. It is simple to use, easy and quick of attachment.

5. Normal adjustment, allowing the extensometer to be used on diameters or thicknesses up to 0·625 in., and up to 0·75 in. in width, or, with attachments, 0·015–0·300 in. diameter wire.

6. 2 in. gauge length positively located.

7. The dial can be rotated until in the most convenient position for observation.

Description. The Lindley extensometer (Fig. 38) consists of a body *A* having a rigid arm *B*. To the upper end of the column is hinged a similar arm *C*. The hinge consists of a wide strip of spring steel, which allows

the arm to move in a vertical plane while preventing any sideways or rotational movement. The specimen under test is gripped between the ends of the screws *E*. These are operated by means of thumb-wheels *F*, through gears housed in casing *G*, in such a manner that the vertical axis of the specimen, whatever the thickness of the latter, is always coincident with the central plane of the extensometer. To the upper arm of the extensometer is secured a spring-steel lever *H*, which passes down the front of the body and carries a cone-shaped button. Pressure on the lever forces the cone into a hole at the base of the column, so arranged that when this is done the distance between the gripping screws is 2 in. (3 in. on some models).

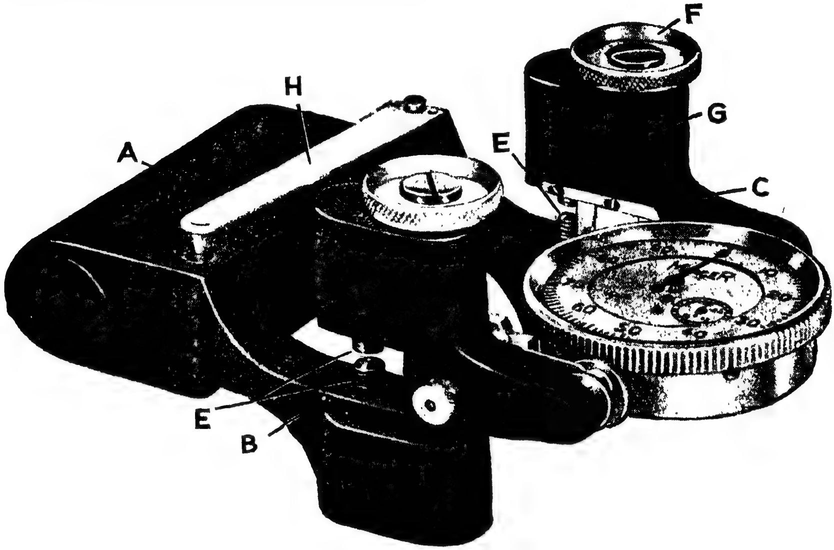


FIG. 38. LINDLEY EXTENSOMETER
(*J. E. Baty & Co.*)

The outer end of each arm is provided with a hardened steel bush, recessed to take one of the ball ends with which the special Mercer dial gauge is fitted. The ends are retained in the sockets by means of forked springs, which prevent end play but permit the dial to be rotated into any convenient position for observation. The maximum extension of which the extensometer is capable is 0·1 in. on the specimen (0·2 in. on the dial), and the instrument should be removed as soon as sufficient extension for the purpose of the test has been recorded. In the event, however, of undue elongation or premature fracture of the test piece, the dial is released without damage, and although the spring hinge may be strained this is easily removed for correction or replacement.

Normally the weight of the instrument may be ignored, but if the specimen is of very small cross-section, the weight of the extensometer can be taken into consideration. If the testing machine is of the type in

which the pull on the upper grip is measured, the machine should be balanced with the grips in position before inserting the specimen and extensometer. The latter has been designed with the movable arm uppermost, in order that the weight of the instrument is carried by the specimen under test and included in the pull measured by the beam. If the testing machine is so arranged that the pull on the lower jaw is

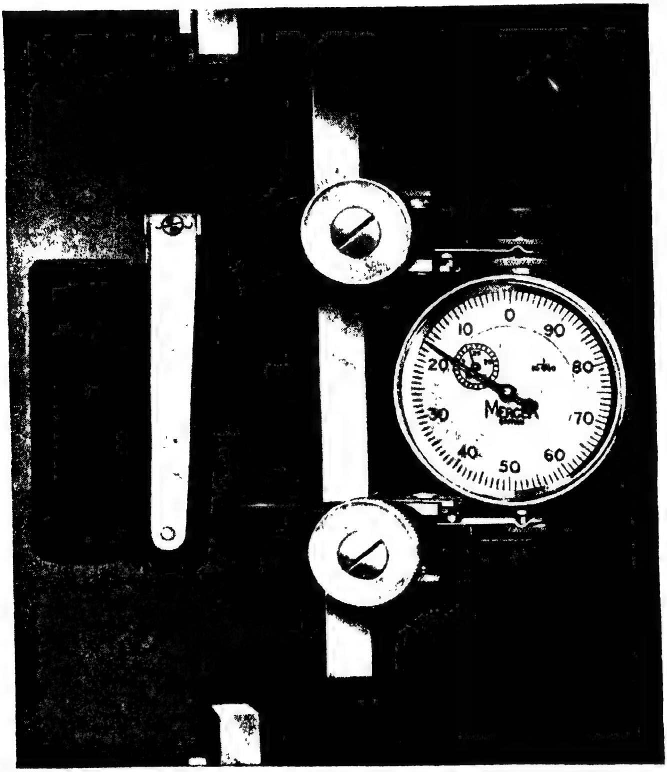


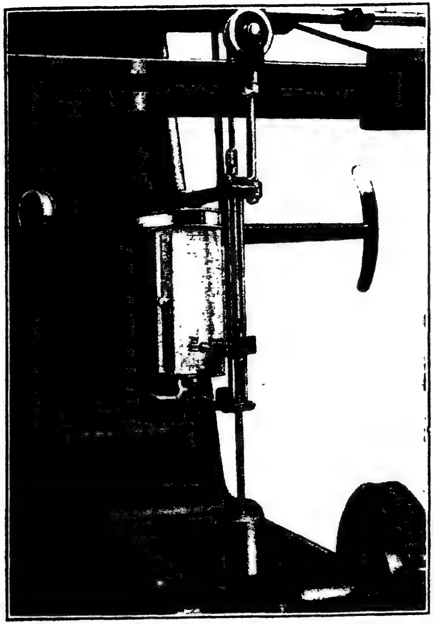
FIG. 39. LINDLEY EXTENSOMETER IN USE

measured, the weight of the extensometer should be added to the reading obtained.

All extensions for test results given in this book were measured by the "Lindley" extensometer.

AUTOGRAPHIC RECORDERS

For record purposes on routine testing, and when accurate measurement of extension is not required, it is often useful to have a load/extension graph of the complete test, plotted by the machine itself. This is accomplished by autographic recorders, which, for the single-lever and



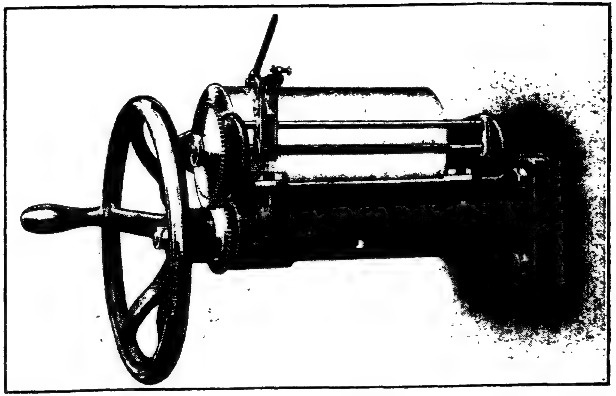


Fig. 40. Autographic Recorders
(*Avery*)

multiple-lever type machine, are of two main types, according to the method adopted for registering the load. These methods of load recording are—

1. From the deformation of a calibrated spring.
2. From the travel of the poise weight along the lever,

With both types the extension ordinate is obtained from the deformation of the test specimen.

Both types are illustrated in Fig. 40, and type I is seen fully set up in Fig. 17, which also shows the calibration springs fitted between the end of the lever and a mounting on the end support.

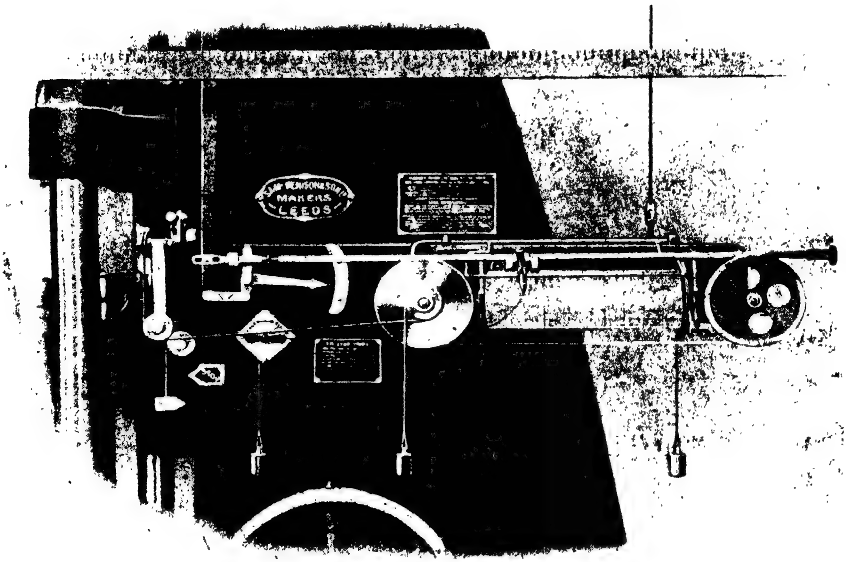


FIG. 41. DENISON DRUM TYPE AUTOGRAPHIC RECORDER ON SINGLE LEVER MACHINE

1. The Spring-loaded Type. With this in use, the poise weight is moved along the beam to a position in excess of the maximum load for the test specimen, the beam being supported above the lower stop by the calibrated spring. The particular spring calibration depends upon the maximum load used. When the straining gear is operated, applying load to the specimen, the spring is relieved of the same load and contracts. The reduction in length of the spring is proportional to the load, and is indicated on the drum of the recorder by means of a steel tape passing over guide pulleys and actuating the pencil in a vertical plane.

Extension of the test piece is recorded horizontally by rotation of the drum. This rotation is effected by another tape, passing round suitably

arranged pulleys, and in connection with the top and bottom tension holders.

It should be noted that, with this type of recorder, the poise weight is not traversed along the lever during a test, and it is therefore only necessary to arrange for a satisfactory straining speed for the test to be automatically carried out. The operator has only to set up, start, and

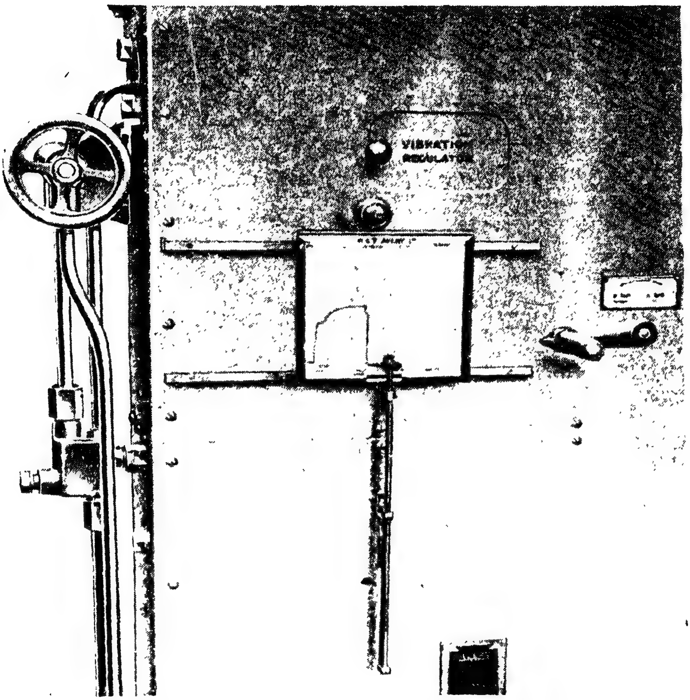


FIG. 42. AUTOGRAPHIC RECORDER ON AVERY SELF-INDICATING UNIVERSAL MACHINE (TYPE 723/1472)

Showing: Pinion driving rack (above graph plate), horizontal movement indicating deformation of specimen; Graph plate; Barograph pen moving in accordance with load (calibrated spring type); Pen rod, racked for engagement with spur wheel driven from shaft inside load cabinet.

stop the test. The only indication of the test results will be on the plotted graph.

2. The Geared Type. In this, the load ordinate is produced from the travel of the poise weight along the lever and the Avery recorder (Fig. 40) is attached to the bracket carrying the poise-propelling handwheel. The test is carried out in the usual manner by balancing the load through moving the poise, and therefore a visual check on the load scale can be made apart from the recording drum.

This type of recorder is more accurate since it works off the lever and dead weight system of the machine.

The Denison recorder, illustrated in Fig. 41, is used for both types of operation, i.e. calibrated spring or poise weight travel. For single-lever machines compensating mechanism compensates for fluctuations of the steelyard within about plus and minus three per cent of the full capacity of the machine.

On the self-indicating type machines the load registration can be effected in various ways according to the particular design. On the Amsler machine the load registration is taken via the pendulum of the dynamometer, the rod actuating the pointer pinion of the dial indicator being extended to carry the pen, this being shown in Fig. 26B.

Flat Board Recorder. This type of recorder has replaced that of the drum type on the latest Avery and Denison machines. It has the advantage that the whole of the load extension curve is visible during the process of a test, whereas with the drum type chart carrier only a small portion of the curve is visible.

As with other types, the load registration may be effected either through the medium of calibrated springs or from the dead weight system of the machine according to the design.

In the Avery type illustrated in Fig. 42, the extension of the specimen is indicated by horizontal movement of the chart, the load ordinate being produced by vertical movement of the pen.

In the Denison type, which is illustrated fitted to the machine in Fig. 30, the pen carriage is traversed laterally across the frame to produce the extension ordinate, the load ordinate being produced by vertical movement of the chart frame. The chart frame moves in accordance with the load as it is connected to the final weighing member of the lever system of the machine.

CHAPTER IV

HARDNESS TESTS

THE following hardness tests relate to the resistance that a material offers to indentation.

THE BRINELL HARDNESS TEST

In 1900 J. A. Brinell, a Swedish engineer, presented a paper to the Swedish Society of Technologists, dealing with his new method of hardness determination.

The Brinell hardness test is performed by pressing a hardened steel ball into the material under a definite load, the hardness number being obtained by dividing the applied load (in kilogrammes) by the spherical area of the indentation (in square millimetres)—

$$\text{Hardness number} = \frac{P}{\frac{\pi D}{2}\{D - \sqrt{(D^2 - d^2)}\}}$$

where P = Applied load (kg.),

D = Diameter of ball (mm.),

d = Diameter of indentation (mm.).

British Standard Specification No. 240 states that the spherical area must be calculated from the average diameter of the impression, and not from the depth.

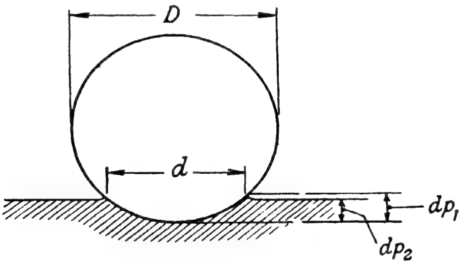


FIG. 43. BRINELL IMPRESSION, SHOWING DEPTH RELATIONSHIP

In the formula above, the depth of the indentation is given by—

$$\text{Depth} = \tfrac{1}{2}\{D - \sqrt{(D^2 - d^2)}\} \text{ mm.}$$

and it might be wondered why a depth-measuring instrument could not be used to solve most of the equation immediately. Reference to Fig. 43 will show the reason.

When a Brinell impression is made, the material adjacent to the

depression is often raised above the level of the surface, in the form of a lip or rim. The reading of the diameter d, corresponding to the depth dp_1, is that used for calculation of the spherical area. If a depth-measuring instrument were used, the depth dp_2 would be recorded, i.e. the depth relative to the normal surface, and this value substituted for dp_1 would not give a true Brinell hardness number; also, if measurements are made after removal of the load, the depth is affected to a much greater extent than the diameter by any elastic recovery of the material.

The raised lip differs with different materials, so that there is no common factor by which dp_2 could be multiplied to give dp_1. For standard Brinell testing, therefore, diameter d must be measured.

Originally, the Brinell test was confined to the use of a 3000 kg. load, using a 10 mm. diameter steel ball. The size of the impression thus obtained confined the application of the test to large pieces, both as regards surface area and thickness. In order that the test might be extended to permit the use of smaller size ball indenters, giving much smaller impressions, and also to make the test comparative for all materials, it is necessary to know the relationship between the load applied and diameter of ball used. Taking steel as an example, if the load is varied on a 10 mm. diameter ball, commencing with the standard load of 3000 kg. and decreasing, it will be found that with *hard* material the hardness number will decrease as soon as the load is decreased. That is, by using different loads on the same size ball, a different hardness number is obtained even on the same material.

The reason is as follows. With varying load the diameter of the impression will vary, and the relation between this diameter and the spherical area will not be constant. In other words, the form of the impression will vary.

In order to obtain accurate hardness numbers it is essential that the indentation be of constant form, i.e. according to the hardness of the material, the load will have to be varied until the impression is of some specified diameter. For example, if a piece of medium-carbon steel is first normalized, then a hardness test made using a 3000 kg. load on a 10 mm. diameter steel ball, the hardness number would be, say, 170 Brinell. Now, if that same piece were hardened and tempered to be much harder than before, and an accurate comparison with the original hardness were required, it would be necessary to increase the load until the impression was the same diameter as before. That is, the two impressions would be identical.

The same would be true if, for example, a piece of brass were to be hardness-tested. In this case, such a load would have to be applied as again would produce an impression of the same diameter as obtained on the steel. Then a true comparison could be made of the relative hardnesses of the steel and the brass. Such tests are impracticable for commercial testing, as in order to know what load to use it would be necessary to know the hardness—and it is the hardness that the test is intended to find. The same difficulties would be experienced when tests using smaller balls had to be made.

In order, then, to overcome the impracticability of a constant impression, the form is allowed to vary within certain limits. The limits

are that the diameter of the impression should not be less than 0·25 or more than 0·5 of the ball diameter, i.e.—

$$d/D = 0{\cdot}25\text{–}0{\cdot}5$$

Fig. 44 is constructed so as to conform to these limits, and shows the variation in the form of the impression.

It can therefore be taken that the ideal form of impression would be the mean of the limits, i.e. $d/D = 0{\cdot}375$. These limits enable tests to be made on steel of varying hardness, using the 10 mm. ball and the same load of 3000 kg., and the hardness numbers obtained are closely approximate.

There is still the question of using smaller ball indenters. If a piece of material is of uniform hardness, the hardness number given by using a smaller ball should be the same as that given by a 10 mm. ball. What load, then, must be used on the smaller ball to give this result?

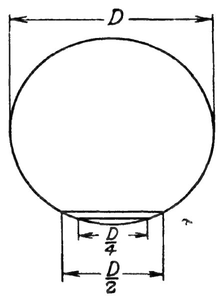


FIG. 44. BRINELL IMPRESSION, SHOWING d/D RATIO.

Various investigators have proved that the principle of geometrical similarity could be applied to ball indentations. This principle has been defined as follows: "With balls of different size, loads proportional to the squares of the ball diameters produce geometrically similar indentations in uniform material and give the same hardness number."

For example, if P = load in kilogrammes and D = diameter of ball—

$$P = D^2 \times \text{constant}$$

or

$$\frac{P}{D^2} = \text{constant}$$

The constant in the above expression, upon which depends the actual magnitude of the load, is chosen according to the average hardness of particular material. Also it must be of such a value that—

$$d/D = 0{\cdot}25\text{–}0{\cdot}5$$

In order, therefore, that Brinell hardness tests may be standardized, both as regards ball diameters and loads, the British Standards Institution specify particular sizes of balls and certain values for the constant.

The P/D^2 values for representative materials are—

Steels, cast iron	30
Copper alloys, aluminium alloys	10
Copper, aluminium	5
Lead, tin, and their alloys	1

and the size of ball indenters standardized are 1, 2, 5, and 10 mm.

The application is as follows. If a 5 mm. diameter ball is to be used on steel, then load P is given by—

$$P/5^2 = 30 \quad \text{i.e. } P = 750 \text{ kg.}$$

If a 10 mm. ball is to be used on brass or duralumin—

$$P/10^2 = 10 \quad \text{i.e. } P = 1000 \text{ kg.}$$

The law of geometrical similarity is borne out thus. If a 10 mm. ball and a 3000 kg. load on steel gives an impression diameter of 3·6 mm., then a 750 kg. load on a 5 mm. diameter ball will give an impression of 1·8 mm., i.e. $d/D = 0{\cdot}36$ in each case.

Also the spherical area for the 1·8 mm. impression will be one-quarter that of the 3·6 mm. diameter, so that if the load is reduced to one-quarter of its previous value of 3000 kg. the spherical area of the resulting impression given by 750 kg. will also be one-quarter, thereby giving the same hardness number, e.g.—

$$H = \frac{3000}{\text{spherical area}} = \frac{750}{(\text{spherical area})/4}$$

TABLE XI. STANDARD LOADS FOR BRINELL HARDNESS TESTS

Ball dia.	Load (kg.)			
	$P/D^2 = 30$	$P/D^2 = 10$	$P/D^2 = 5$	$P/D^2 = 1$
mm.				
1	30	10	5	1
2	120	40	20	4
5	750	250	125	25
10	3000	1000	500	100

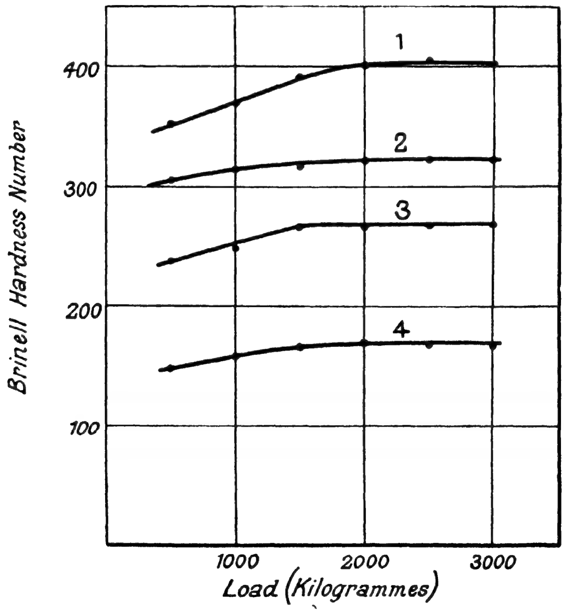


FIG. 45. GRAPHS INDICATING VARIATION IN HARDNESS NUMBER WITH LOAD ON BALL

Variation of Load on the Same Ball. Earlier mention has been made of the variation in hardness number caused by variation of load using the same diameter ball. Reference to Fig. 45 and Table XII shows that this effect is not very pronounced below hardnesses of about 400 Brinell until very low loads are employed. The reason is, that with the permissible limit of $d/D = 0{\cdot}25$–$0{\cdot}5$, within which comparable tests can be made, if the standard load gives such an impression that $d/D =$ say 0·4, then a big reduction in load could be made before the value becomes equal to the low limit of 0·25. Any load which will give d/D between 0·25 and 0·5 will therefore give an approximately correct hardness number, as Table XII shows. As soon as the ratio falls below the low limit, then the hardness numbers fall.

For standard testing it is of course not permissible to vary the load from that given by the appropriate P/D^2 ratio; but, as has been shown, it is not correct to assume that the hardness number varies as soon as the load is varied, except for hard material. Hard material tested with the standard load will give d/D near the low limit directly, and any further reduction of load would naturally give a value lower than 0·25.

This matter of load variation can be also viewed in the following manner. If it is standard to use a 3000 kg. load on a 10 mm. ball for steels of any hardness within the Brinell range, e.g. for a mild steel giving a Brinell hardness of 100 and also for a heat-treated steel of 400 Brinell, then surely it is no different to vary the load on the 10 mm. ball on the *same* material. In *each* case d/D will vary, but so long as the limits of 0·25–0·5 are maintained the hardness numbers will be closely approximate and comparable.

Brinell Tests on Hard Materials. The Brinell test is most useful up to a hardness number of about 400 when using ordinary steel balls, but with special work-hardened balls the useful range is extended to about 650 Brinell. The limit of the useful range is set by the deformation of the ball under load. This deformation causes the indentations to become more shallow and of greater diameter than if the ball remained truly spherical. The hardness numbers thus obtained are lower than the true value, and the harder the material the greater becomes the error. This means that, although a material of 200 Brinell can be said to be twice as hard as one of 100 Brinell, it cannot be said that a material 800 Brinell is four times as hard as material of 200 Brinell. High Brinell numbers have no relation to those much lower in the scale, and thus the former numbers are really meaningless so far as comparison is concerned.

In spite of this, statements may be read in technical publications to the effect that the hardness of such and such a material was increased from say 300 Brinell to 900 Brinell (converted V.P.N.). Such a statement gives the impression that the material has been made three times as hard as previously, which is totally incorrect, whereas a much more accurate comparison would have been made had the original Vickers Pyramid Numeral figure been stated. Why this should be so will be dealt with later, when dealing with the Vickers Diamond Pyramid hardness test. It was for the very reason that the Brinell test proved quite inadequate for very hard materials that other tests using diamond indenters were evolved.

TABLE XII. BRINELL NUMBER WITH VARYING LOAD
(10 mm. Ball)

Material	Load	Indentation dia.	d/D	Hardness No.
	kg.	mm.		
(1) Medium-carbon steel 3S6 Hardened 850° C., tempered 350° C.	3000	3·05	0·305	401
	2500	2·8	0·280	406
	2000	2·5	0·250	400
	1500	2·2	0·220	390
	1000	1·85	0·185	368
	500	1·35	0·135	346
(2) 3S6 Tempered 450° C.	3000	3·4	0·340	321
	2500	3·1	0·310	322
	2000	2·775	0·2775	322
	1500	2·42	0·242	319
	1000	2·0	0·200	315
	500	1·45	0·145	303
(3) Nickel-chrome steel 4S11 Heat-treated	3000	3·7	0·370	269
	2500	3·4	0·340	267
	2000	3·05	0·305	267
	1500	2·65	0·265	267
	1000	2·25	0·225	248
	500	1·65	0·165	232
(4) 3S6 Normalized	3000	4·65	0·465	167
	2500	4·25	0·425	168
	2000	3·8	0·380	169
	1500	3·35	0·335	165
	1000	2·8	0·280	159
	500	2·05	0·205	149

TABLE XIII. BRINELL NUMBER WITH STANDARD LOADS
($P/D^2 = 30$)

Material	Load	Ball dia.	Indentation dia.	d/D	Hardness No.
	kg.	mm.	mm.		
4S11 Heat-treated	3000	10	3·700	0·370	269
	750	5	1·850	0·370	269
	120	2	0·742	0·371	268
3S6 Normalized	3000	10	4·600	0·460	170
	750	5	2·300	0·460	170
	120	2	0·909	0·454	175

($P/D^2 = 10$)

Brass	1000	10	3·450	0·345	104
	250	5	1·750	0·350	101
	40	2	0·686	0·343	105

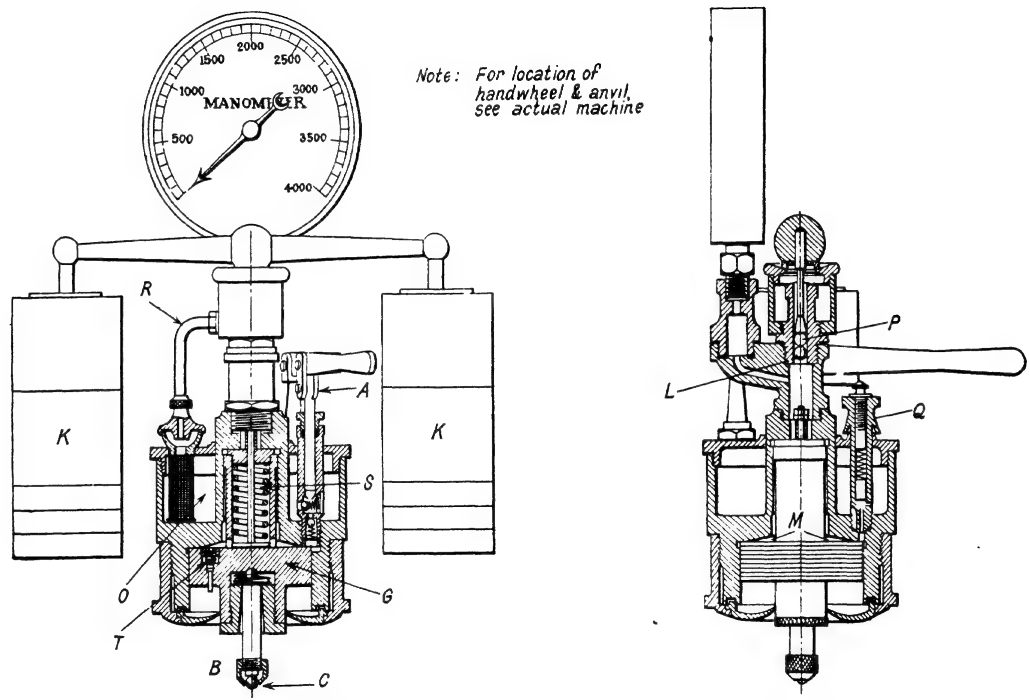


FIG. 46. CROSS-SECTION OF HYDRAULIC PRESS UNIT (BRINELL MACHINE)

THE BRINELL MACHINE

The machine (Figs. 46, 47) consists of a vertical hydraulic press, into the cylinder of which oil is forced from a container *O* by means of a hand

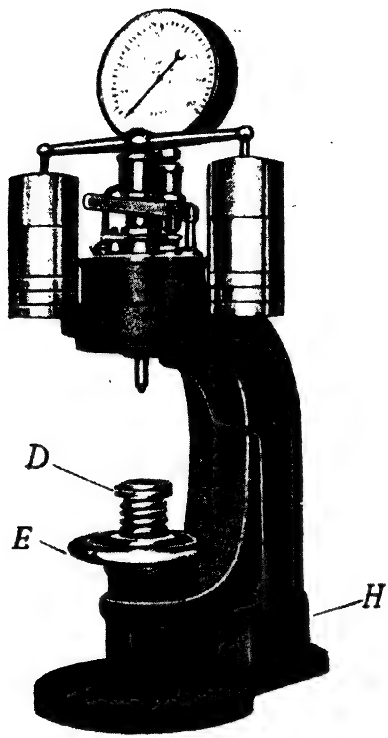


FIG. 47. STANDARD BRINELL MACHINE
(*J. W. Jackman & Co. Ltd.*)

pump *A*. On the lower part of the piston assembly is a ball-holder *B*, to which the ball indenter *C* is fitted.

The specimen is placed on the press table *D*, which is vertically adjustable by means of the handwheel *E*. The press table is spherically seated in the top of the screw so that the plane surface of irregular specimens may be adjusted to a horizontal position.

The ram *G* works without packing, and leakage is prevented by the accurate fitting of the ram into the press cylinder, any small quantity which does pass the ram being collected in the receptacle *H*, from which it is periodically returned to the oil container.

The force exerted by the piston is recorded on a gauge graduated in

kilogrammes, and a deadweight control K is provided which accurately regulates the load required for the test. This controlling device consists of a smaller cylinder L in direct communication with the main cylinder M. A ball piston P in the smaller cylinder is loaded with proportional weights corresponding to the testing load required. When these weights rise, the requisite load is obtained, and maintained, for the duration of the test.

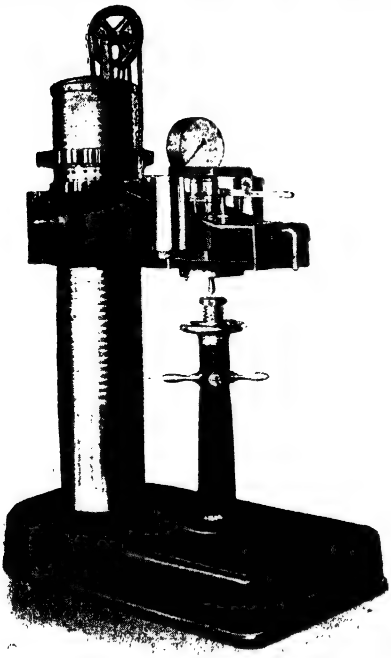


FIG. 48. RADIAL-ARM TYPE MACHINE

(*J. W. Jackman & Co. Ltd.*)

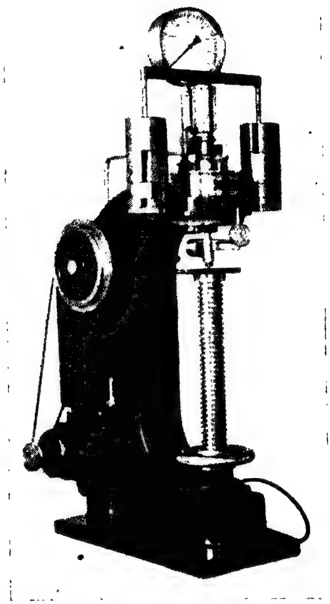


FIG. 49. MOTOR-DRIVEN BRINELL MACHINE WITH DEPTH GAUGE

(*J. W. Jackman & Co. Ltd.*)

Any small leakage of oil past the ball piston is returned to the oil container via pipe R.

The pressure in the press cylinder is released by opening a valve Q, and the spring S, which is a little stronger than the weight of the piston, effects its return. The indentation is then ascertained by means of a measuring microscope, which is graduated in millimetres and can be read accurately to 0·05 mm.

When tests are made in close succession, it may happen that there is not sufficient time for the piston to return to its original position after every test before valve Q is closed for the next test. The piston would be thus gradually forced out of the cylinder, but this is prevented by a valve T in the piston, which opens automatically when the piston has

reached its lowest position and allows the oil to pass through the piston, when it is drained away to an outside receptacle.

This is indicated by the required pressure load not being attained by continued pumping.

FURTHER POINTS IN CONNECTION WITH THE MACHINE AND THE TEST

The Deadweight Control. The action of the deadweight control may be explained by reference to Fig. 50, which illustrates the press cylinder M, ram G, smaller cylinder L, and ball piston P. W illustrates the weight on the piston P.

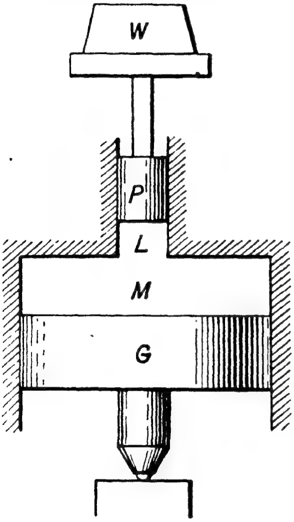


FIG. 50. DIAGRAMMATIC SKETCH OF BRINELL DEADWEIGHT CONTROL

Assume that the area of the top of the main ram is 10 sq. in. and that of the piston P $\frac{1}{4}$ sq. in., and that an oil pressure of 500 lb./sq. in. is being exerted on each.

Then the total force which would be applied to the ball indenter would be $10 \times 500 = 5000$ lb., whereas due to the smaller area of the upper piston the upward force would be $\frac{1}{4} \times 500 = 125$ lb. Therefore, if a weight W of 125 lb. were placed on piston P, it would (neglecting friction) give rise to a pressure of 500 lb./sq. in. to the oil column underneath, which pressure acting on the larger area of the ram would transmit a force of 5000 lb. to the ball indenter, and this force would be maintained as long as W was supported entirely by the oil. This, then, is the action of the deadweight control in the machine.

The smaller piston consists of two 10 mm. diameter steel balls, the centre spindle of the yoke carrying the weights being cup-shaped to fit on the uppermost ball. The use of the ball piston reduces friction to the very minimum.

In carrying out a test, in order to avoid the possibility of overloading due to any stickiness of the ball piston (cold oil, etc.) the operator should assist the yoke to rise and continue pumping until the required load is nearly reached. On approaching the required load stop pumping and gently lower the yoke, when the load corresponding to the equivalent weights on the yoke will be registered.

It will be clear from the explanation that the weights hung on the yoke are not marked with their true weights, but represent the effective load transmitted to the ball indenter by the ram proper.

Due to the small area of the ball exposed to the oil (half the surface area), a small weight can act as a control. Also, with the form of the deadweight control employed, the correct load is maintained as the ram descends during indentation.

The Weights and Balls Supplied. The yoke itself corresponds to a load on the indenter of 187·5 kg. Additional weights corresponding to 62·5,

250, 500, and 1000 kg. are also supplied, there being two of each. For example, for a load control of 250 kg., one weight of 62·5 kg. would be placed on each arm of the yoke.

It has been previously mentioned that for British Standard testing the ball sizes are standardized at 10, 5, 2, and 1 mm. diameter. The balls supplied with the standard machine are of 10, 5, and 2·5 mm. diameter.

Also, the greatest load which can be used on a 2 mm. diameter ball will be on steel, when $P/D^2 = 30$, giving 120 kg. load. Therefore with the standard Brinell machine no 2 mm. ball impressions can be carried out, as the lowest load obtainable is 187·5 kg., which is the load required on a 2·5 mm. ball to continental standards.

The standard Brinell machine is not very sensitive to light loads, and is therefore limited to 10 mm. and 5 mm. diameter ball impressions.

The Load Gauge. This is essentially a pressure gauge, but instead of being calibrated in kilogrammes per square millimetre of oil pressure, the pressure calibrations are multiplied by the area of the top of the ram in square millimetres, thereby giving kilogrammes load.

If during a test, with the weights floating, the load indicated on the gauge does not agree with that on the yoke, the gauge must be at fault. The gauge is intended to indicate the amount of the load as pumping is progressing, thus enabling the operator to avoid overloading.

Reading the Diameter of the Impression. Measuring microscopes with or without artificial illumination are obtainable with the machine. The diameter of the impression is ascertained by graticule reading from a scale in the microscope.

If accurate readings are needed, eyestrain is unavoidable, apart from the difficulty of trying to see two opposite tangents to the circle at the same time, especially on 10 mm. ball impressions.

RELATIONSHIP OF BRINELL HARDNESS NUMBER TO ULTIMATE TENSILE STRESS OF STEEL

An important feature of the Brinell test is that the hardness number so obtained enables a close approximation to be made to the ultimate stress of steel, with the exception of those severely cold worked or special steels. In these latter cases, the relationship must be fixed by experimental results and not by the following ratios.

The following details give the factor by which the Brinell hardness number is to be multiplied in order to ascertain the approximate tensile strength—

Carbon steels	0·23
Alloy steels, heat-treated to 60 tons/sq. in.	0.22
Alloy steels, heat-treated to 100 tons/sq. in.	0·21

PRECAUTIONS TO OBSERVE WHEN CARRYING OUT A BRINELL HARDNESS TEST

1. Make certain that the surface to be tested is adequate for the size of the ball to be used. The B.S.I. state that the centre of the impression must not be less than two and a half times the diameter of the impression

from any edge, and that generally the thickness must be ten times the depth of the impression.

2. As the accuracy of reading the indentation diameter depends upon the surface finish, it is better, when circumstances permit, evenly to polish the surface, or part of the surface to be tested, even when using a 10 mm. diameter ball. Generally, an even polish down to No. 0 emery is very suitable for both 10 mm. and 5 mm. diameter impressions, and No. 00 paper for 2 mm. diameter balls.

3. If the surface has to be ground for preparation, make certain that it is carried out so as not to affect the condition of the material, e.g. rash grinding resulting in overheating.

4. See that the surface to be tested is square to the ball holder, otherwise an oval impression will result.

5. Elevate the work against the ball until the ram is pushed upwards.

6. Pump evenly and assist the yoke to rise when the load is within about 250 kg. of that required.

7. Do not continue to pump when the load is at the required value. If the weights are barely floating, lift them by hand and then pump, then gently lower the weights.

8. Fifteen seconds is the time now standardized for the application of the load for all values of P/D^2.

9. Read the impression in two places at right angles, and take the mean of the reading.

10. When expressing the hardness number, state also the ball diameter used and the load applied, e.g. H 10/3000 = 285.

THE VICKERS DIAMOND PYRAMID HARDNESS TEST

Previous explanation relative to the Brinell test has demonstrated that apart from the errors involved due to ball deformation, etc., the hardness numbers so obtained are not at all comparative, i.e. a material of Brinell hardness of 700 cannot be taken as twice as hard as a material of 350 Brinell, and in order that comparative hardness tests could be made on hard material, beyond the range afforded by the Brinell test, the Vickers Diamond Pyramid test was evolved.

The Vickers method of hardness determination is an ingenious adoption of the principle of geometrical similarity of impression based on the Brinell type of test.

In order that approximate comparisons of hardness can be made within the range of Brinell testing, it is necessary that the ratio d/D = 0·25–0·5. The mean of these limits would therefore give an ideal form of impression, i.e. $d/D = 0{\cdot}375$.

Fig. 51 is so constructed that the diameter of the impression d is actually 0·375 of D. At the points of contact of d with the circle, tangents are drawn forming the angle of indentation, which for this construction is 136°.

It is clear that if the ratio of d/D has any other value than 0·375, then the angle of indentation will vary. If it were possible to keep the angle of indentation constant, then an ideal Brinell impression would always result and the hardness numbers be accurate. This being the case, why not utilize the angle for the purpose of indentation? The angle must, of

course, be incorporated in some finite form, e.g. a cone or a pyramid, the angle between opposite faces being the 136°.

The Vickers machine, by utilizing a square-base pyramidal diamond, not only solves the question of constant angle of indentation and geometrical similarity, but also the question of deformation of the indenter under load. Moreover, the hardness number obtained is truly in accordance with the Brinell principle and is calculated in the same manner, i.e.—

$$\text{Diamond hardness number} = \frac{\text{Load (in kg.)}}{\text{Area of indentation (sq. mm.)}}$$

$$= \frac{2\,P \sin \theta/2}{d^2}$$

where P = Load (kg.)

d = Diagonal of impression (mm.),

θ = Angle between opposite faces of pyramid.

With constant angle of indentation and no measurable distortion under load, the Vickers diamond test gives a hardness scale which is truly proportional, i.e. a material of 700 hardness will be twice as hard as one having a hardness of 350, etc. This fact is of great importance, as it provides a rational scale of hardness, from the softest to the hardest material. This feature is not possessed by any other method of hardness test, except those employing constant-angle diamond indenters, and expressing the hardness number as load/area.

It should be noted that any other angle than 136° as used in the Vickers test would give a proportional scale. The angle of 136° is used on account of its relationship to the ideal form of Brinell impression as shown in Fig. 51. On this account, Vickers diamond hardness numbers are true numbers, and the two scales agree until the varying factors inseparable from the Brinell test cause increasing divergence as the hardness increases.

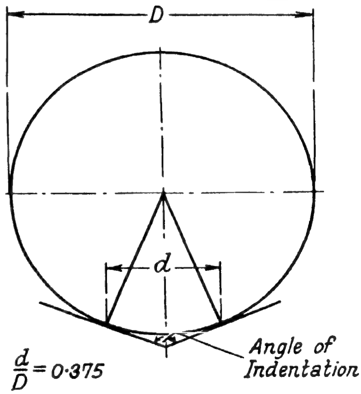


FIG. 51. ANGLE OF INDENTATION FROM IDEAL BRINELL IMPRESSION

Hardness Number Constant with Variation of Load. In order to make a smaller impression than that given by a 10 mm. ball in the Brinell test it is necessary to use a smaller standard ball, the load applied being given by the particular P/D^2 ratio. This is in order to keep the angle of indentation within the limits d/D = 0·25–0·5.

As the Vickers diamond system gives a constant angle of indentation irrespective of the load applied, it follows that the hardness number will also be constant whatever the load. If the load is doubled, for example, then the area of the indentation will be twice as great, giving the same hardness number. The load can therefore be chosen according to require-

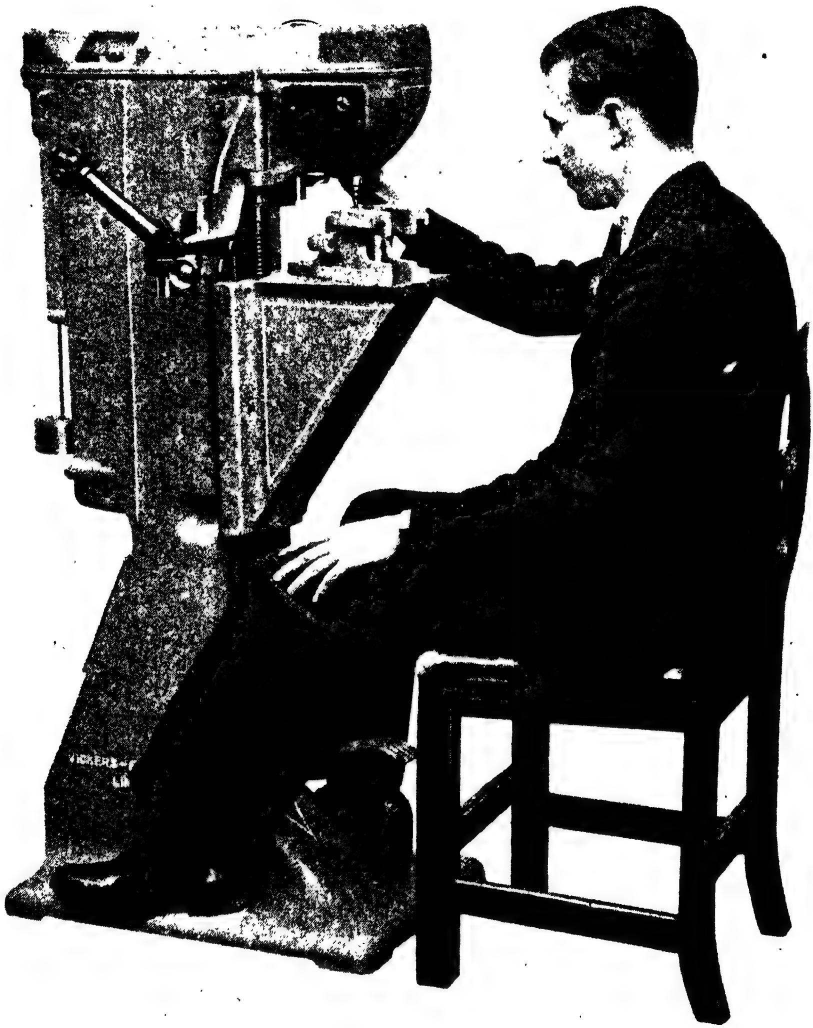


FIG. 52. THE VICKERS DIAMOND PYRAMID HARDNESS TESTER
(Vickers-Armstrongs Ltd.)

ments. If a very thin piece of material is to be tested, then a light load giving a small impression is required. Larger and harder material can have corresponding heavier loads. There is therefore no P/D^2 ratio for the Vickers diamond test.

Although generally no variation of hardness number is obtained when the load is varied on homogeneous material, some materials may give a variation. This is due to a work-hardening or, in some cases, to a work-softening effect as the load is increased. For standard testing it is therefore necessary to state the particular load which was used when obtaining

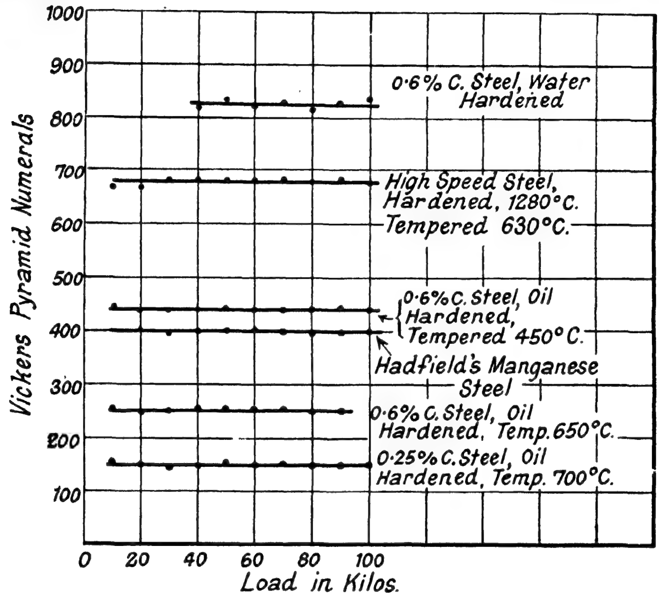


FIG. 53. GRAPH SHOWING HARDNESS WITH VARYING LOAD, USING VICKERS DIAMOND MACHINE

a hardness number, and further, for some aircraft material, one specified load is indicated (e.g. 10 kg. for duralumin). The load used should be indicated in all test reports, e.g. HD/10 = 118.

Fig. 53 gives the results obtained in several tests.

The Standard Loads. For standard testing to B.S.S. No. 427, the loads used must be 5, 10, 20, 30, 50, 100, and 120 kg. wherever possible.

It is also stated that in order to obtain the greatest accuracy the load should be as large as possible. This load will of course be governed by considerations of the material and its dimensions. The use of the largest possible load obviates errors due to any variation between the surface layer and bulk of the material, and the larger the area of indentation the less will be the percentage error in reading the diagonal. The following table shows how greater accuracy is obtained by using heavier loads on hard material.

TABLE XIV

Ocular reading	Hardness numeral									
	0	1	2	3	4	5	6	7	8	9
(a)* 250	297	294	292	289	287	285	283	281	279	276
(b)* 560	296	295	294	293	292	291	289	288	287	286
(a)* 130	1097	1081	1064	1048	1033	1018	1003	988	974	960
(b)* 300	1030	1023	1016	1010	1003	997	990	984	978	971

* (a) 10 kg. load ⅔ in. objective; (b) 50 kg. load ⅔ in. objective.

It will be seen that although the hardness numerals can be read to approximately the same accuracy on the softer material, using the 50 kg. load enables a much closer determination to be made when handling the harder material.

Test Made to Show that the Vickers Diamond Hardness Scale is Proportional. The tabulated results indicate that the tool steel is $817/268 = 3{\cdot}049$ times as hard as the 4S11 nickel-chrome steel. If this is correct, then by applying a load of $30 \times 3{\cdot}049 = 91{\cdot}47$ kg. to the tool steel, the same area of indentation should be obtained as for the 4S11 with the 30 kg. load.

Material	Load	Ocular reading	Hardness No. (V.P.N.)
	kg.		
4S11	30	455	268
Tool steel, hardened .	30	261	817

A 91·5 kg. load (the nearest available) was then applied to the tool steel with the following result—

Material	Load	Ocular reading	Hardness No. (V.P.N.)
	kg.		
Tool steel, as above .	91·5	452	816

It is seen how closely the practical result is to that anticipated.

Reading the Impression. The square form of impression given by the Vickers diamond test has decided advantages. In the first place, measurement across the diagonals of the impression is more simple than reading the diameter of an indentation given by a spherical or conical indenter. Secondly, the "draw" or "cratering" which takes place when an indentation is made is confined to the sides of the pyramid and is absent from the corners. Thus measurement takes place at points which are free from these effects, and the resulting error is confined to the difference in

area between the actual impression and the true square. This is a negligible amount. Thirdly, the method of direct measurement by callipering between two knife-edges is infinitely more easy, more accurate, and abolishes the eyestrain associated with graticule reading as in the Brinell microscope.

Eyestrain is abolished by vertical illumination from within the measuring microscope, which causes the impression to appear black against an illuminated background (Fig. 54). The knife-edges of the callipers are also black, and measurement of the diagonal is automatically recorded on a counter (Fig. 55). By turning the head of the microscope through 90° the other diagonal may be measured, and if there is any variation the mean of the two will be the true reading.

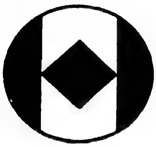


FIG. 54 MEASUREMENT OF VICKERS DIAMOND IMPRESSION

The microscope reads to 0·001 mm. (approximately 1/25,000 in.) when using a $\frac{2}{3}$ in. objective, and to 0·0025 mm. when using a $1\frac{1}{2}$ in. objective.

Another great feature connected with the reading is that the microscope positions itself immediately over an impression, and focusing is the only operation necessary in order to obtain the impression in the centre of the field of vision.

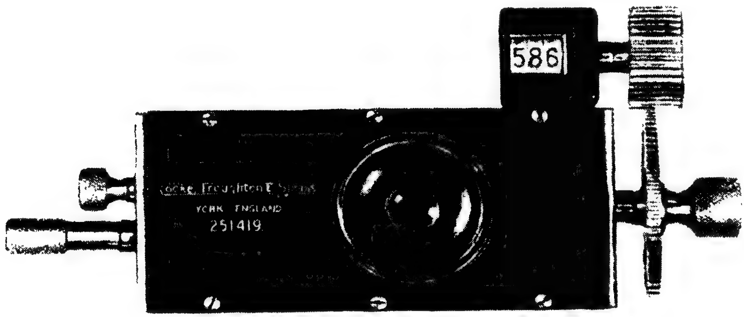


FIG. 55. OCULAR MICROMETER
(*Vickers-Armstrongs Ltd.*)

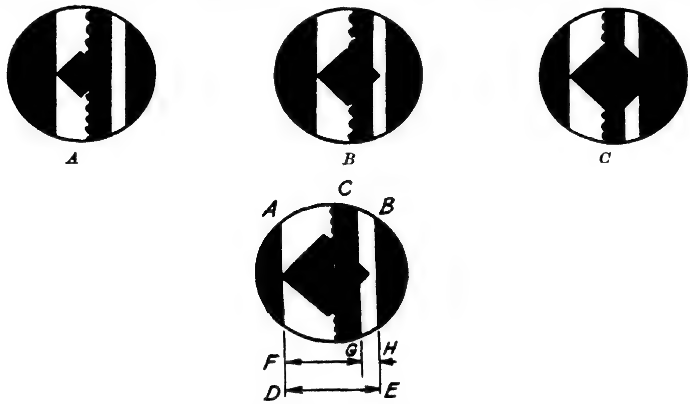
Uniform Loading. In an automatic machine, the rate, smoothness of loading, and duration of load are independent of the operator.

As the loads used in the Vickers test are very small in comparison with the Brinell, the Vickers machine is direct lever loading and automatically controlled. The load is gradually applied by a cam, maintained for a given interval, and then released. The cam rotation is governed by a dashpot, the control valve of which can be adjusted so that the time taken in the application and duration of the load can be varied according

to requirements. The load should be maintained for 15 sec. for Standard testing requirements.

Application for Brinell Testing. The Vickers machine can be employed as an accurate light-load Brinell tester by substituting either a 1 mm. or 2 mm. ball and holder in place of the diamond unit. In this case the loads used must conform to Brinell standards. The diameter of the indentation can be read very accurately and easily by means of the callipering arrangement previously mentioned.

For Checking Minimum and Maximum Limits. Often for routine work a hardness test is only carried out to see that the hardness of the material is within a specified range. To obviate the delay in callipering each impression individually a third knife-edge (Figs. 56 and 57) is utilized, by



FIGS. 56 AND 57. MEASUREMENT OF HARDNESS WITHIN LIMITS

A shows the material too hard.
B shows the material correct.
C shows the material too soft.

means of which the limits of hardness are defined. In Fig. 57 *A* and *B* are the normal knife-edges, and *C* is brought into use for maximum and minimum limit testing. Knife-edge *C* has a serrated left-hand edge in order that it should not be confused with its right-hand side, which is used for the check.

The operation of setting the knife-edges is as follows. *B* is adjusted so that *FG* equals the diagonal reading which would give the maximum limit of hardness. *C* is then adjusted so that its measuring edge is just touching the knife-edge of *B*. *B* is then screwed back until *DE* represents the diagonal reading of minimum hardness.

When measuring any impression, *A* is set against the left-hand corner of the diagonal. Knife-edges *B* and *C* will move with *A* and keep their relative positions. Therefore, if the material is to conform to the limits, the right-hand corner of the diagonal of the impression must be between *G* and *E*.

[*Note.* When returning the auxiliary knife-edge *C* after limit checking it is most important that it be screwed fully back into the left-hand side.

If it is left slightly projecting past A it cannot be seen, as both knife-edges are black. Therefore, when A is adjusted to the left-hand edge of an impression, in reality it will be the right-hand edge of C which will be viewed, and a totally incorrect subsequent reading of the diagonal will result. To avoid this, it is a good plan to check the zero. The right-hand

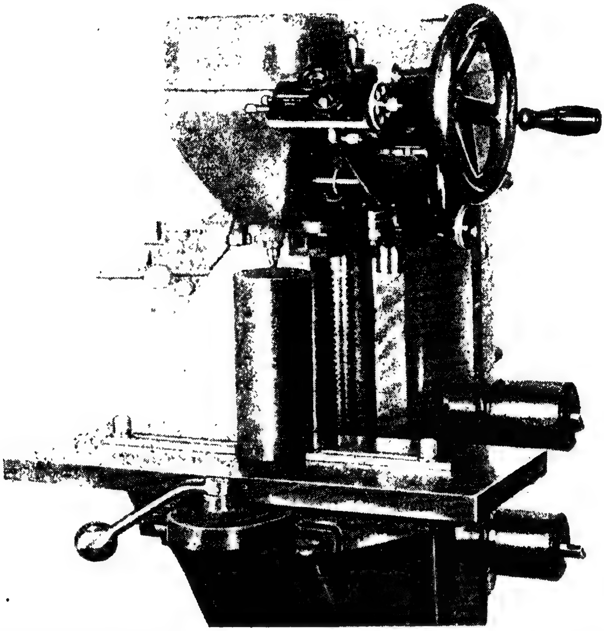


FIG. 58. SLIDING-TABLE ATTACHMENT FOR VICKERS DIAMOND MACHINE
(*a*) Making the impression.

knife-edge B is brought up against A, and when the edges just meet the ocular reading should be 000.]

Sliding Table Attachment. When using the machine ordinarily it is necessary to raise the stage to make an impression, and to lower it in order that the microscope may be swung over for measuring. The use of the sliding table attachment obviates the necessity for raising and lowering the stage. In this case the microscope is fitted on the right-hand side of the machine at such a level that when the work is moved after an impression has been made the minimum of adjustment for focusing is necessary. The operation of the sliding table is shown by Figs. 58 and 59.

SUMMARY OF POINTS RELATIVE TO THE VICKERS DIAMOND PYRAMID HARDNESS TEST

1. The hardness numbers obtained are recognised as standard.
2. The one machine covers all requirements.

3. The hardness scale is proportional.
4. Uniformity of load and duration.
5. No limit to hardness of metals that can be tested.
6. No damage to finished work, as the impressions are very small.
7. Applicable to very thin sheet.
8. Accuracy and ease of readings.
9. Rapid checking to limits.
10. Applicable to light-load Brinell testing with 1 mm. and 2 mm. diameter balls.
11. All working parts are completely enclosed.

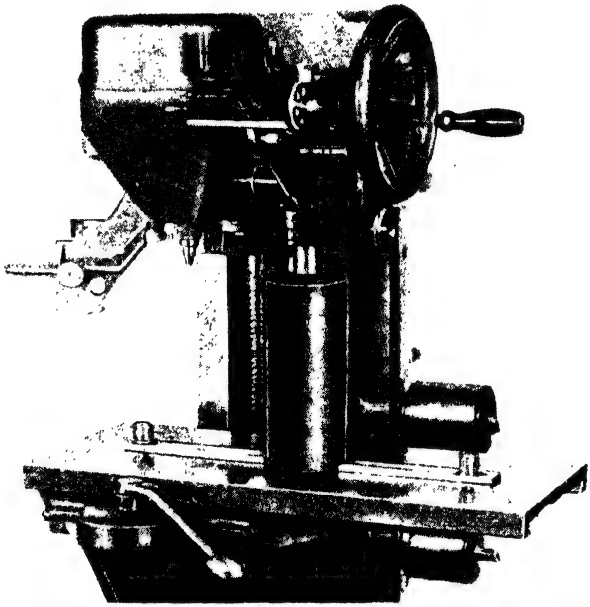


FIG. 59. SLIDING-TABLE ATTACHMENT FOR VICKERS DIAMOND MACHINE
(b) Measuring the impression.

Description of the Vickers Diamond Hardness Testing Machine. The machine, which is shown diagrammatically in Fig. 60, consists of a main frame *F* of U-section, which carries the stage *S* and a simple lever *L* of 20 : 1 ratio, applying the load through a thrust rod *Tr* to a tube *T*, which is free to reciprocate vertically and carries a diamond indenter *D* at its lower end.

Attached to the main frame is a smaller frame *Fm*, which contains all the control mechanism. The plunger *Pl* reciprocates vertically under the influence of a rotating cam *C*, its purpose being to apply and release the test load. The cam is mounted on a drum, and when the starting

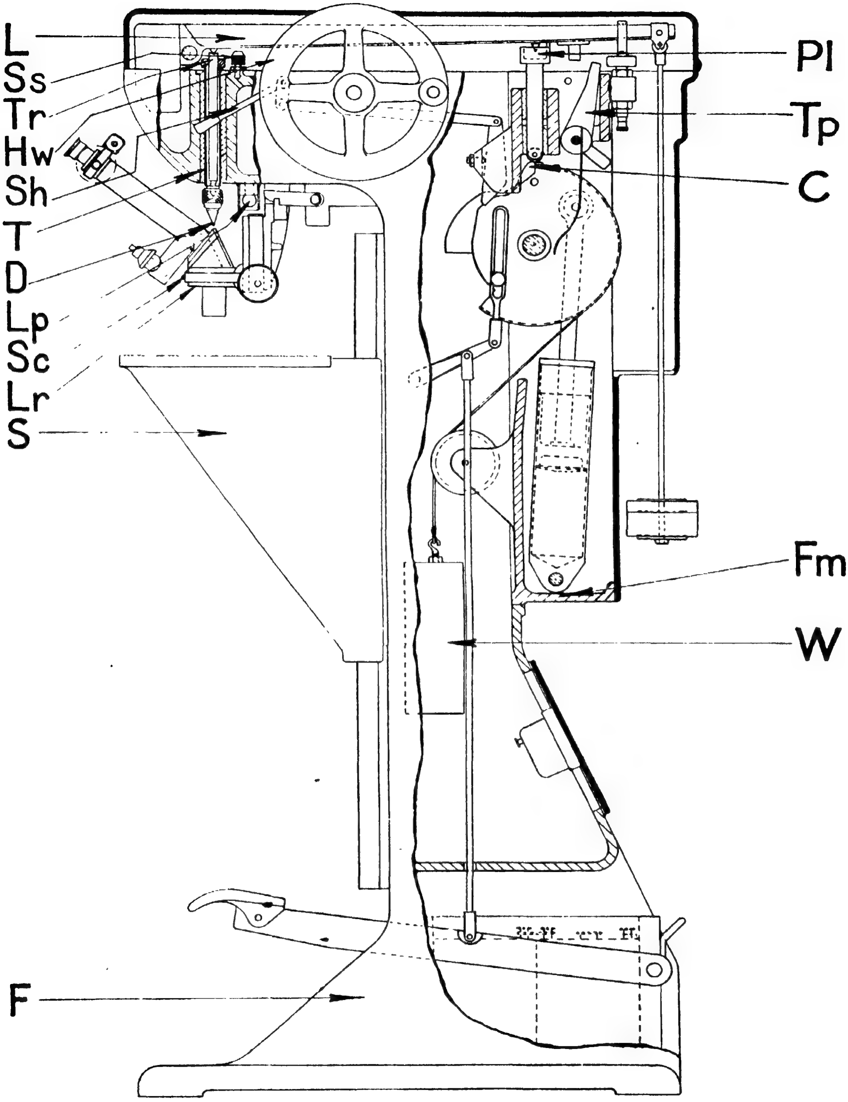


FIG. 60. CROSS-SECTION DIAGRAM OF VICKERS DIAMOND MACHINE

handle *Sh* has been depressed, the whole is rotated by a weight *W* attached by a flexible wire, the speed of rotation being controlled by a piston and dashpot of oil. The rate of displacement of the oil is regulated by an adjustable control valve. The plunger carries a spring pad at its upper end which engages with a cone mounted in the beam, thereby ensuring a very slow and diminishing rate of application for the last portion of the load. Since the cam both lowers and raises the plunger, it will be seen that uniformity of loading and duration of load are attained, all errors due to inertia and premature removal of the load being eliminated.

Depression of the foot pedal returns the cam, drum, and weight to their original positions. A tripping piece *Tp* supports the beam during this latter operation, and drops out as soon as the plunger returns to its top position. The machine is then ready for another test.

The machine is supplied with weights ranging from 1 kg. to 120 kg. in order that the load most suitable to the piece under test may be employed.

Testing Instructions. The weight rod is loaded with any of the weights supplied to suit requirements. For normal purposes a load of 30 kg. is applied, but in certain cases, e.g. when testing cast iron or non-ferrous castings and the general hardness is required, it is desirable to use a heavier load in order to average up possible heterogeneity. For this purpose a load of 100 kg. is recommended.

When testing thin material, care must be taken to select a load which is not too heavy for the piece under test. The British Standards Institution recommends that the thickness of the test specimen shall be at least equal to one and a half times the diagonal of the impression. (See Fig. 61.)

It has been found that, owing to the critical nature of very light loads, too small a gap left between the diamond indenter and the work under test vitiates the hardness figures, and in this respect it is strongly recommended that operators form the habit of leaving a gap of at least ½ mm.

For case-hardened work it is often of great assistance to judge the useful depth of the case by applying diminishing or increasing loads until the hardness figure remains steady. The pedal should first be depressed in order to load the machine. The stage is then raised until the surface to be tested just clears the point of the diamond. The small lamp *Lp* may be switched on if necessary, to facilitate this adjustment. The starting handle *Sh* is then pressed, when the test proceeds automatically and terminates with an audible click. If for any reason the work has not been elevated sufficiently, a small buzzer comes into operation and warns the operator that the test is not being carried out correctly.

The stage is then lowered until the microscope can be swung over the work, the impression focused, and the measurement made as follows. The left-hand knife-edge is adjusted by means of the lower knurled thumb-screw on the left-hand side of the microscope headpiece, so as to correspond to the left-hand corner of the impression, and the right-hand knife-edge, controlled by a micrometer screw connected to a counting mechanism on the right-hand side of the headpiece, is moved to correspond to the right-hand corner of the impression. The reading is taken

from the figures shown on the side of the eyepiece and converted by means of tables supplied direct to Vickers Pyramid Numerals (V.P.N.).

If both diagonals of the impression require to be measured, the headpiece is fitted with a stop which enables it to be instantly rotated through 90°.

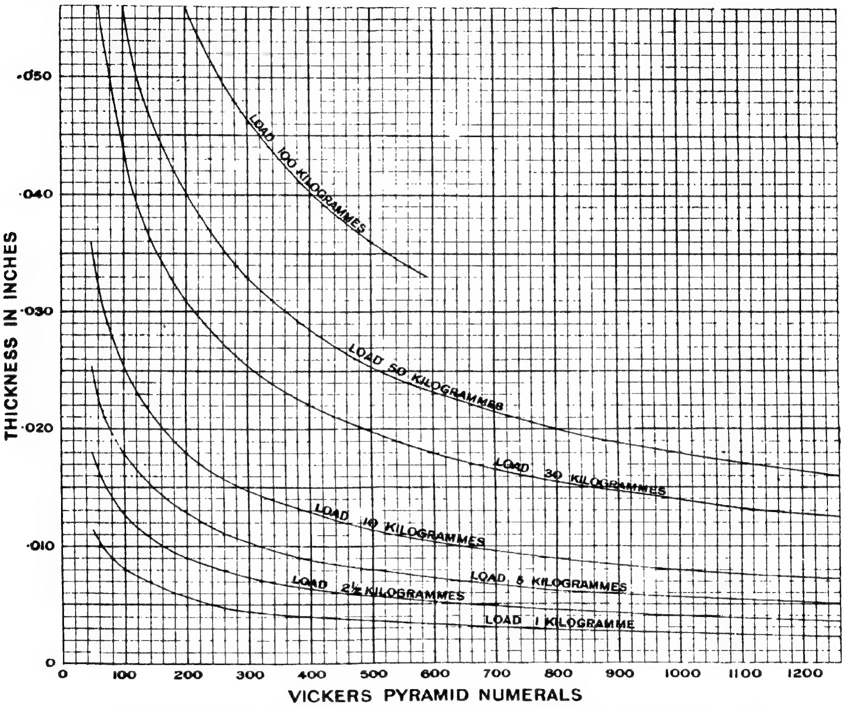


FIG. 61. CURVES SHOWING MINIMUM THICKNESS OF MATERIAL USING VARIOUS LOADS

(*Vickers-Armstrongs Ltd.*)

Vickers Diamond and Brinell Hardness Numbers. As Vickers Diamond hardness numbers are obtained from a test in which the angle of indentation remains constant, and Brinell numbers with a varying angle, that is, between the limits $d/D = 0{\cdot}25 - 0{\cdot}50$, it is evident that in most cases there will be a difference in the hardness numbers given by the Vickers and the Brinell test when testing the same homogeneous material.

Unless a Brinell impression is such that $d/D = 0{\cdot}375$, equivalent to an indentation angle of 136 deg., the Brinell number will differ from that given by the Vickers test, even assuming that there is no deformation of the Brinell ball.

If d/D is less than 0·375, then the Brinell number will be less than the Vickers, and if greater than 0·375 then the Brinell number will be in excess

of the Vickers. For example, if when testing duralumin sheet a 40 kg. load on a 2 mm. dia. ball gives a Brinell hardness number of 110, the Vickers Diamond number using a 10 kg. load on the same material would be 115. The reason for this difference is due to the fact that when using a standard load of 40 kg. ($P/D^2 = 10$) on duralumin, the impression does not conform to $d/D = 0{\cdot}375$, but is less.

If, however, the load on the 2 mm. ball is increased above 40 kg. until an impression diameter of 0·750 mm. is obtained, giving $d/D = 0{\cdot}375$, then the Brinell number will be exactly the same as the Vickers, namely 115.

The following test results are given to illustrate the effect. It should be noted that the Brinell tests all conform to $d/D = 0{\cdot}25 - 0{\cdot}50$, although the actual British Standard load is 40 kg.

Material	Test.	Load kg.	Diagonal or diameter of impression mm.	d/D (Constant for diamond ·375)	Hardness No.
Forged Duralumin Bar	Vickers	10	·385	·3750	125 V.P.N.
Spec. 6 L1	Brinell (2 mm. ball)	40	·645	·3275	119 Brinell
		56	·745	·3725	124·7 Brinell

It will therefore be evident that no accurate general conversion of Vickers to Brinell numbers, or vice versa, can be made unless graphs be prepared for each material or each P/D^2 ratio under consideration.

Conversion tables can be only approximate owing to the varying angle of indentation of the Brinell ball, even on different materials giving the same hardness number, i.e. when using different values of P/D^2.

CHAPTER V

HARDNESS TESTS (CONTINUED) AND IMPACT TESTS

THE ROCKWELL HARDNESS TESTER

THIS machine (Fig. 62) is direct-reading, i.e. hardness values are directly indicated on a dial, the value obtained being based upon depth of penetration of an indenter. Instead of the total depth of penetration being used,

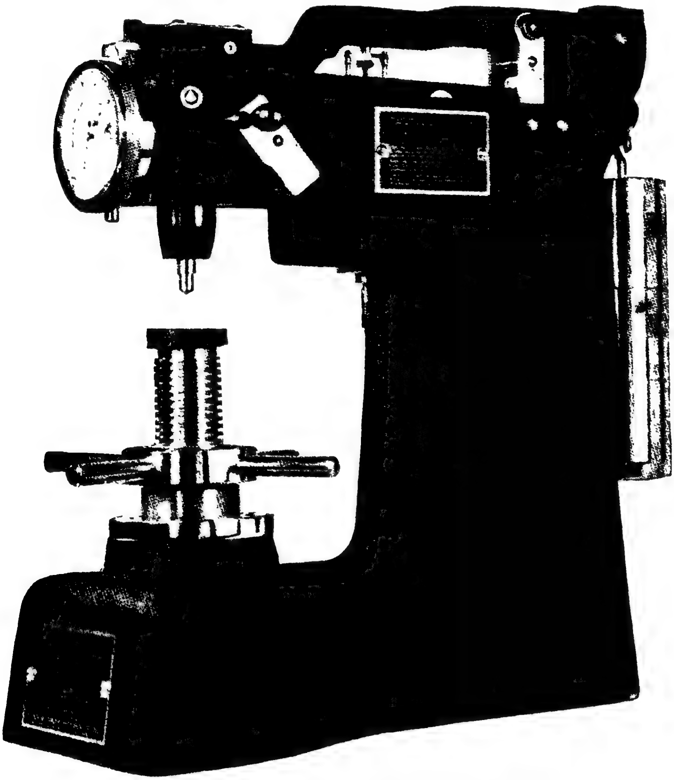


FIG. 62. ROCKWELL STANDARD MACHINE
(George H. Alexander Machinery Ltd.)

the hardness number is based upon the increment of depth which takes place when a second load is applied. The initial load is called the minor load, and the sum of the initial and subsequent loads, the major load.

The principle of increment in depth instead of measurement of total depth or total diameter was adopted—

1. to obtain readings more independent of surface conditions,
2. to obtain readings without confusion through the deformation round the indentation, and
3. to make precision testing possible at a rapid rate without loss of accuracy.

The load is applied by deadweights and lever, and an adjustable oil dashpot controls the rate of application. The dial is calibrated with two scales, one, the *B* scale, used for testing with a $\frac{1}{16}$ in. diameter ball indenter, and the other, the *C* scale, used when testing with a 120° sphero-conical diamond.

The standard loads used for the two indenters are 10 kg. minor load and 100 kg. major load for the $\frac{1}{16}$ in. ball, and 10 kg. minor load and 150 kg. major load for the diamond. Other loads as indicated in Table VIII can be used for special purposes. The $\frac{1}{16}$ in. diameter ball indenter is for use on unhardened steels, cast iron, bronze, brass, etc., while the diamond is for hardened steel.

This diamond has a cone angle of 120° with a spherical tip ground to a radius of 0·2 mm., and is marketed under the trade name "Brale." On hardened steel, using Brale indenter, the total depth of impression varies from 0·0025 in. to 0·0035 in., and on softer steels with the Brale or on soft brass, etc., with the steel ball, the deepest indentation is about 0·010 in., so that the impressions do not mar the appearance of the work and will not cause failure of stressed parts.

Generally, no particular surface preparation is necessary, and ordinary machined surfaces are quite as satisfactory as polished surfaces. The increment of depth due to increment of load is 0·00008 in. for each point of hardness on the Rockwell scale. If a piece of steel is as hard as Rockwell *C*58 at one spot and *C*55 at another, it means that the depth of penetration is 3 × 0·00008 in., or 0·00024 in. deeper at the softer spot.

Making a Test. 1. Place the part upon the anvil of the elevating screw.

2. Turn the capstan handwheel, bringing the work into contact with the indenter, and continue elevating until the small pointer on the dial stands approximately at the dot. (See Fig. 63.) This shows that the initial minor load of 10 kg. has been applied, and the main pointer on the dial will now be approximately vertical.

3. Turn bezel of gauge to set dial zero behind the pointer. The dial zero is marked by a small arrow.

4. Push back crank handle to release and apply the major load. The dial needle will then move in an anti-clockwise direction.

5. Observe when the pointer comes to rest, and then—

6. Pull handle forward, thereby removing the major, but not the minor load.

7. Read Rockwell hardness number on the dial.

The diagrammatic sequence of operations is indicated in Fig. 64.

An important factor in the consistency of Rockwell readings is that both when the dial is set at zero and when the final reading is taken, the frame of the machine is always under the same stress, which is the stress

due to the 10 kg. minor load. Depth readings under the varying loads are absolutely avoided.

Any elastic recovery of the metal upon reduction of major load to minor load does not enter into the hardness reading.

The Dial Calibration. To conform to accepted practice the dial (Fig. 63) is so calibrated that a shallow indentation, meaning hard material, gives a high number, and low numbers mean softer material.

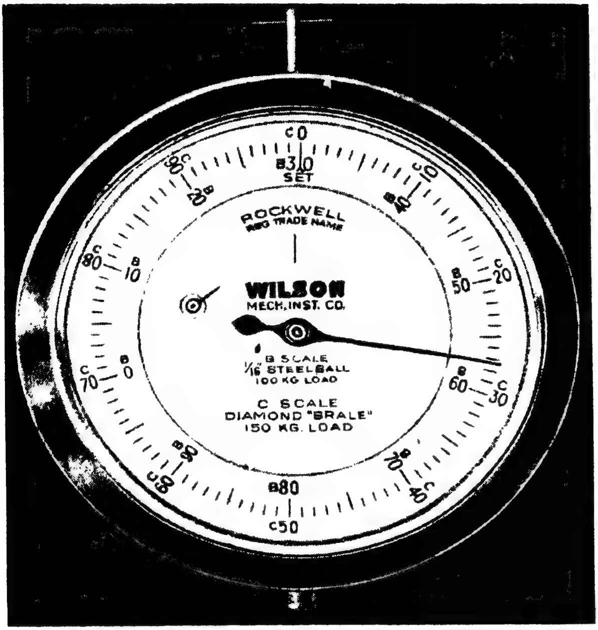


FIG. 63. DIAL ON ROCKWELL MACHINE

This is accomplished as follows. If the dial were calibrated directly in terms of increment of depth measurement, the harder the material the less would be the penetration, and therefore the lower the reading of the dial; but this would not be in accordance with the customary interpretation of hardness numbers as given by the Brinell, etc., where the harder the material the greater becomes the hardness number.

Therefore, in order to conform to accepted practice, the Rockwell dial calibration is effected in this manner. A material of infinite hardness would not indent at all, but instead of giving this a value of 0 it is given on the *C* scale a value of 100. Now, imagine that another material indents to a depth of 10 scale units. Compared to the number 100 assigned to infinitely hard materials its relative number will be 100 — 10 = 90. In this way the harder the material the higher the hardness number on the scale.

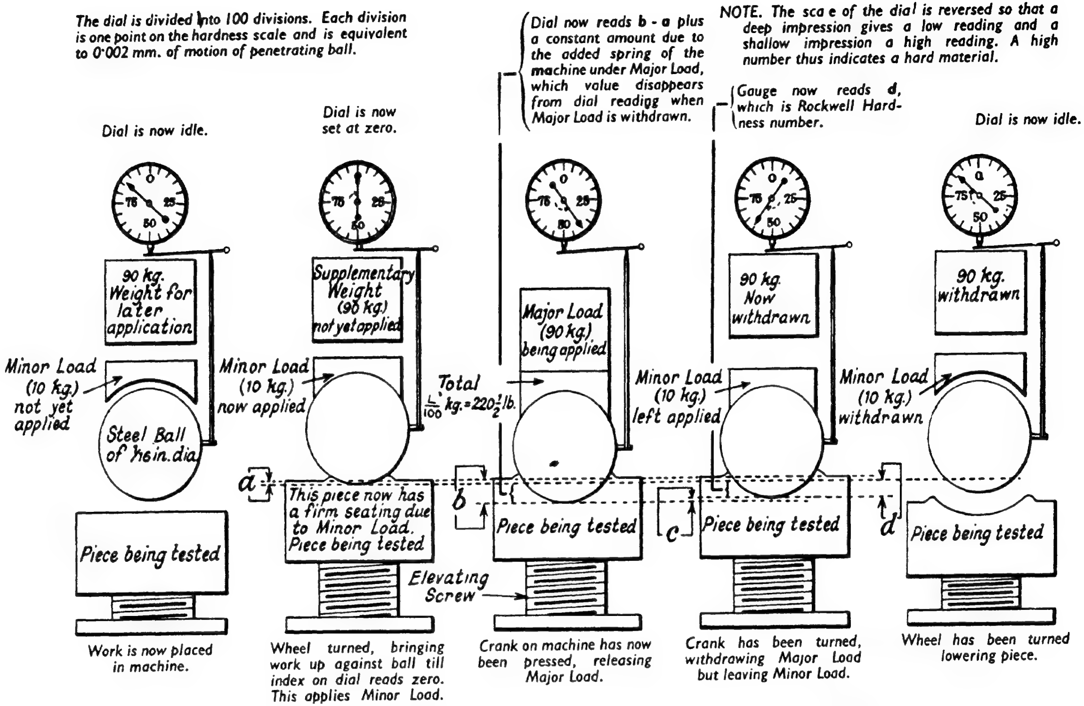


FIG. 64. DIAGRAMMATIC SEQUENCE OF OPERATIONS FOR ROCKWELL MACHINE

In order to illustrate the principle, the size of the ball is greatly exaggerated.

a = Depth of impression made by Minor Load of 10 kg
b = Depth of impression after Major Load of 100 kg. has been applied.
c = Recovery of metal upon reduction of Major to Minor Load. This is an index of the elasticity of metal under test, and does not enter the hardness reading.
d = Difference in depth of holes made = Rockwell Hardness number.

It is therefore seen that if the actual increments of depth due to the major load are required, the Rockwell *C* scale reading must be subtracted from 100, then multiplied by 0·00008 in., which represents the penetration depth of 1 scale division. For example, Rockwell *C*65 is equal to an increment of depth of—

$$(100 - 65) \times 0{\cdot}00008 = 0{\cdot}0028 \text{ in.}$$

In other words, the Rockwell *C* scale number equals 100 minus the actual depth increment, i.e.—

$$C65 = 100 - 0{\cdot}0028 \text{ in.}$$
$$= 100 - 35 \text{ scale divisions}$$

The reason that the zero of the *B* scale is moved 30 points to the left as compared with the *C* scale is to avoid readings below zero on soft brass, etc., which would require a negative symbol if *B* zero and *C* zero coincided.

For metals softer than *C*20, use the $\frac{1}{16}$ in. ball and 100 kg. load. For metals harder than *B*100, use Brale diamond and 150 kg. load.

TABLE XV. STANDARD AND SPECIAL ROCKWELL SCALES

Scale	Penetrator	Major load	Dial numbers	Remarks
		(kg.)		
B	$\frac{1}{16}$ in. **Ball**	**100**	**Red**	
C	**Brale diamond**	**150**	**Black**	
A	Brale	60	Black	
D	Brale	100	Black	
E	$\frac{1}{8}$ in. Ball	100	Red	
F	$\frac{1}{16}$ in. Ball	60	Red	
G	$\frac{1}{16}$ in. Ball	150	Red	
H	$\frac{1}{8}$ in. Ball	60	Red	
K	$\frac{1}{8}$ in. Ball	150	Red	

Accessories. A variety of ingenious jigs and anvils is available for testing rounds, rings such as outer races of bearings, etc., and for light and heavy work of long overhang. (See Fig. 65.)

A special gooseneck adapter for the indenter permits testing the inner cylindrical surfaces of many tubes and annular parts, and if a large tube or cylinder has to be tested from 1 in. to 6 in. in from one end, a special internal tester is used.

THE ROCKWELL SUPERFICIAL HARDNESS TESTER

This machine operates on exactly the same principles as the regular Rockwell but with much lighter minor and major loads, and it has a more sensitive depth-measuring system. It is designed especially for making hardness tests with very minute impressions applicable to thin metal sheet, lightly carbonized or nitrided steel, etc.

Bearing in mind that any sort of penetration hardness test is influenced by the hardness of the metal tested to a depth of approximately ten times the actual depth of penetration, it will be evident that thin metal or thin, hard cases can accurately be measured only by shallow indentation.

The "Superficial" tester applies a minor load of only 3 kg. and major

loads of 15, 30, and 45 kg.. Whereas hardened tool steel is penetrated to a depth of about 0·0025–0·0035 in. on the standard Rockwell *C* scale test, the "Superficial" tester, on metal of the same hardness and with the

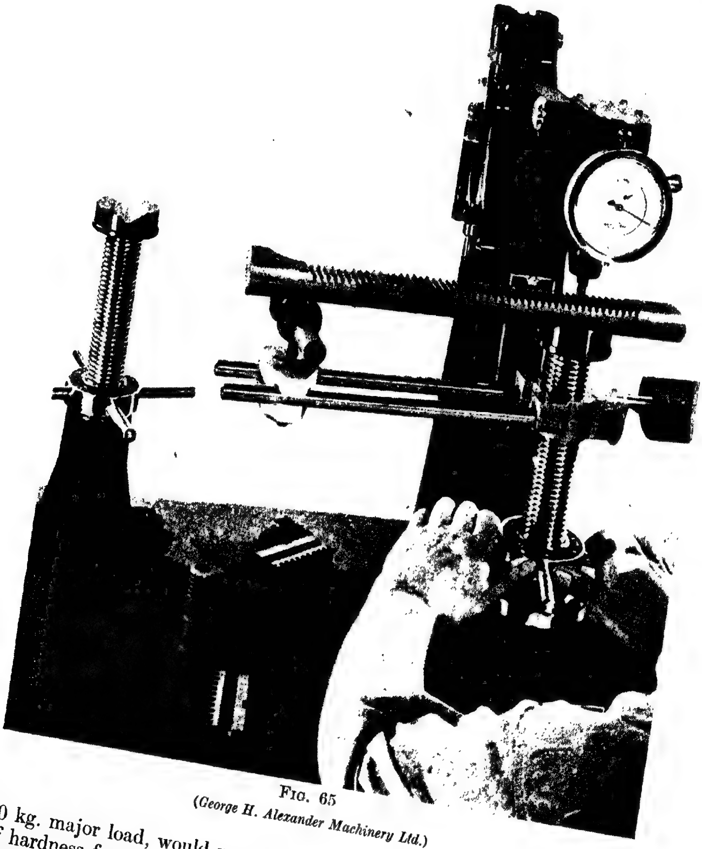


Fig. 65
(George H. Alexander Machinery Ltd.)

30 kg. major load, would penetrate to only about 0·0010 in. One point of hardness for any test made on a "Superficial" tester corresponds to difference in depth of penetration of only 0·001 mm. (approximately 0·00004 in.). In recording readings obtained on this model, the major load in kilogrammes is first stated, then the letter *N*, if Brale *N* indenter is used, or the letter *T* if $\frac{1}{16}$ in. diameter ball, and finally the dial reading.

The Brale N indenter used on this machine is a special precision type of the ordinary Brale used on the larger machine.

GENERAL

As Rockwell hardness numbers are based on depth measurement with constant load, the accuracy of determining absolute hardness values decreases with increasing hardness. In other words, a difference in scale reading of 1 unit (0·00008 in.) will indicate a much greater variation in hardness when dealing with hard material than the variation caused by 1 unit on softer material. This, of course, means that the scale is not proportional. For example, a Rockwell $C40$ is approximately equal to a Vickers diamond of 385, and Rockwell $C41$ equal to V.P.N. 390. However, $C60$ = V.P.N. 746, and $C61$ = V.P.N. 775.

Taking the Vickers diamond scale as truly proportional, it is seen that in the first place one unit difference on the Rockwell C scale is equal to 5 V.P.N., whereas with harder material one unit C scale is equal to 29 V.P.N. These conversion figures are only approximate, but they illustrate that the Rockwell scale cannot differentiate small differences in absolute hardness when dealing with hard material. For purpose of checking uniformity of material for industrial use, however, the direct-reading dial has obvious advantages.

THE AVERY DIRECT-READING HARDNESS TESTER

This machine works on similar principles to the Rockwell, the dial graduations, loads used, and indenters being also the same as for the latter machine.

THE STANDARD FIRTH HARDOMETER

The principle embodied in the Firth Hardometer for determining the hardness of metals is similar to that of the Brinell machine. For hard material a pyramidal diamond indenter, made to the standard angle of 136° between opposite faces, is used.

The hardness numbers are obtained as with the Brinell and Vickers diamond pyramid machines, by dividing the applied load in kilogrammes by the surface area of the indentation in square millimetres.

For the measurement of impressions, necessary for the calculation of area, a microscope is used, and the makers give the following reasons for the adoption of the microscope in preference to the use of a direct-reading dial indicator—

1. The direct-reading dial indicator reading is governed by the depth of impression, which is only from one-eighth to one-sixteenth of the diameter or diagonal of the impression.

2. Since the magnification of the scale of the microscope remains constant, it is more reliable than a mechanical measuring device which is called upon to function on each reading, and, on account of the wear of the moving parts, eventually becomes inaccurate and loses sensitiveness. The importance of the latter point will be appreciated when it is remembered that the depths of the impressions are exceedingly small, and in the case of the hardest steel may be as small as 0·0005 in.

Another feature which could be stated is that when using the diamond indenter the hardness scale is proportional, which is not the case with a direct-reading dial indicator, as previously explained.

The Load Mechanism. The load is applied through a specially calibrated spring and patented trip mechanism, and in such a manner that only the exact load can be applied. The whole of the mechanism is entirely enclosed and protected.

Referring to Fig. 66, a trip mechanism automatically arrests the motion

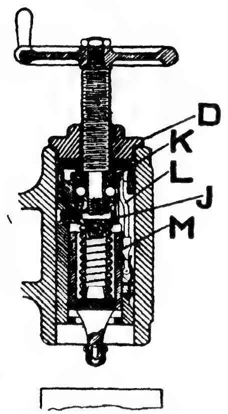


Fig. 66. Load Cylinder (Firth Hardometer)

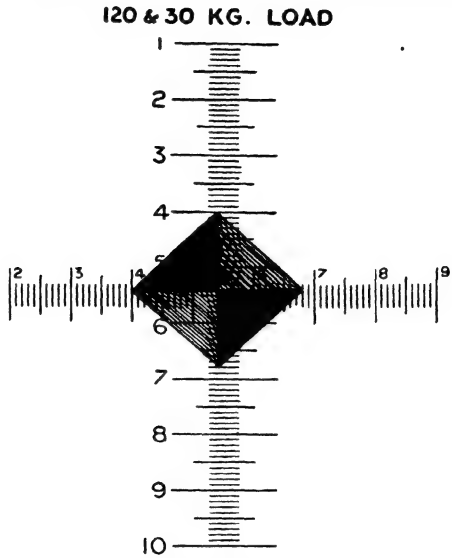


Fig. 67. Micro-projection Head Scale (Firth Hardometer)

of the handwheel as soon as the correct load has been applied. By means of the handwheel and screw, the load cylinder is forced down, carrying with it the trigger M. This trigger, engaging the recess in the inner bushing, immediately allows the stop L to engage the ratchet wheel K.

Load cylinders of 120 kg., 30 kg., and 10 kg. are available.

Calibration. By means of the adjustable spigot J provision is made for calibrating the load. By unscrewing the nut D the load cylinder may be withdrawn entirely from the machine body and, if desired, returned together with the measuring microscope to the maker for recalibration. The setting of the adjusting spigot is sealed before leaving the maker's works.

For the purpose of checking the constancy of the load, a standard check piece of known hardness is supplied with each machine.

The Measuring Microscope. The measurement of impressions is made by means of a microscope fitted with a scale. As with all measuring microscopes with graticule scale, eyestrain results if in continuous use,

and in order to overcome this disadvantage the Firth microscope can be fitted with a projector head which fits over the eyepiece.

The interior of the head is provided with a plane mirror which reflects an enlarged image of the impression on to a hooded ground-glass screen, which is fitted in the front vertical face of the head.

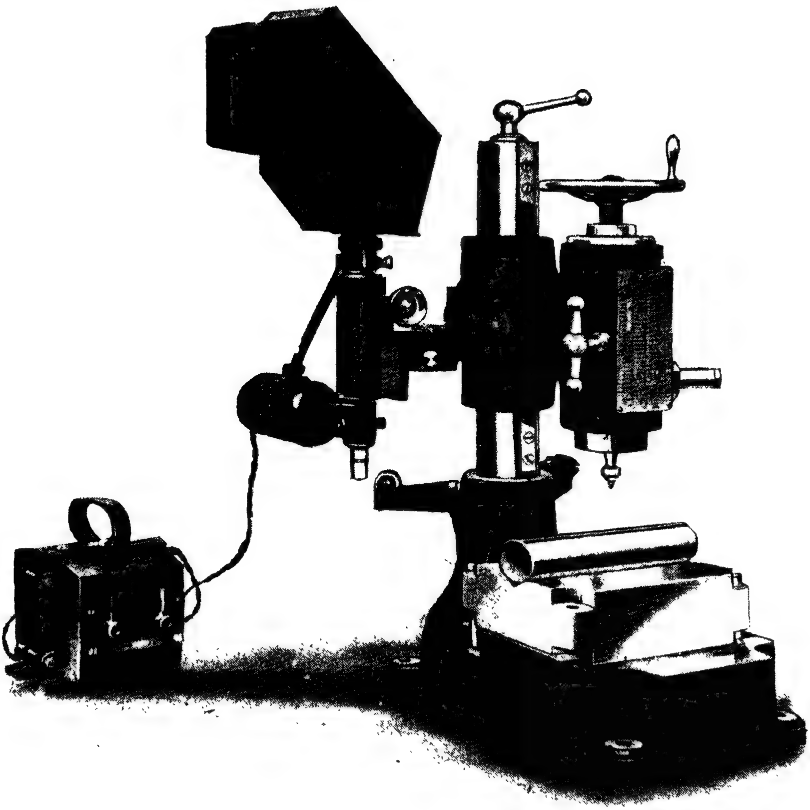


FIG. 68. FIRTH HARDOMETER

The screen is marked with accurately divided cross-scales, an actual-size reproduction being illustrated in Fig. 67. By this means, the operator can read the impression when comfortably seated, use both eyes in an unstrained position, and the greater ease of reading enables the process to be speeded up.

As may be seen from the illustrations, the whole apparatus is of a

simple character, and manipulation of the outfit is well within the competence of the average workshop operator.

For the 120 kg. load cylinder the microscope objective is 1 in. and the smallest scale division is 0·02 mm. With a 30 kg. load cylinder each scale division is 0·01 mm., and for the 10 kg. load 0·005 mm.

The Loads and Indenters. As it is not possible to vary the load on these standard machines, the particular load cylinder must be chosen according to the class of work handled. This is especially important when using ball indenters, which are supplied in 1 mm., 2 mm., and 4 mm. diameter, if the standard ratios of P/D^2 are to be adhered to.

If the machine is to be principally used for testing steel up to 450 Brinell, a 120 kg. load cylinder with a 2 mm. ball would be the correct combination. For harder material the 30 kg. cylinder using the diamond would be necessary. The 10 kg. load cylinder using a diamond indenter is for use with very thin material or material having a very thin carbonized or nitrogen-hardened case.

Making a Test. A small flat should be prepared on the surface to be tested and polished with No. 0 emery.

1. Swing the head into position against the stop (Fig. 68), and lock the column by means of the clamping lever.
2. Support the article on the anvil in a solid manner and bring the ball or diamond into contact with the prepared surface by means of the handwheel.
3. Continue turning the handwheel, which after making about one more revolution will be arrested. The load will then have been applied.
4. Maintain the load for the requisite period, and then withdraw the indenter by turning the handwheel in the opposite direction.
5. Free the column by means of the clamping lever, and swing round until the microscope comes into position immediately over the impression. A stop screw locates the microscope accurately.
6. Adjust the microscope by the thumb-screws until the scale is over the impression. The impression diameter or diagonal may be read in two positions at right angles by turning the eyepiece of the microscope.

It is recommended that when applying the load the handwheel be turned at the rate of one revolution in about ten seconds, and that a downward pressure is not exerted by the hand.

THE FIRTH-BROWN VARIABLE-LOAD HARDOMETER

While the standard Firth hardometer is a very sound instrument, it suffers from the disadvantage of having a fixed load. While the fixed load is quite satisfactory for control purposes on a range of similar materials, the testing of materials of widely varying hardness, and sizes down to a few thousandths of an inch in thickness, necessitates the use of a variable-load machine. To meet these requirements the new variable-load tester has been developed. It incorporates the whole range of testing at present covered by the 10 kg. and the 30 kg. standard machines, and in addition the range is extended.

The load range of this new machine is from 2 kg. to 40 kg., the former allowing the testing of razor-blade strip of 0·003 in. thickness, while the

latter load enables the instrument to conform to the standard $P/D^2 = 10$ ratio, when using a 2 mm. ball for the hardness testing of duralumin to A.I.D. requirements. In addition to these loadings, any intermediate load may also be applied, the full range being indicated on a dial graduated in kilogrammes.

An improved microscope has been fitted to the variable-load machine and is capable of measuring impressions to 0·001 mm., while an additional

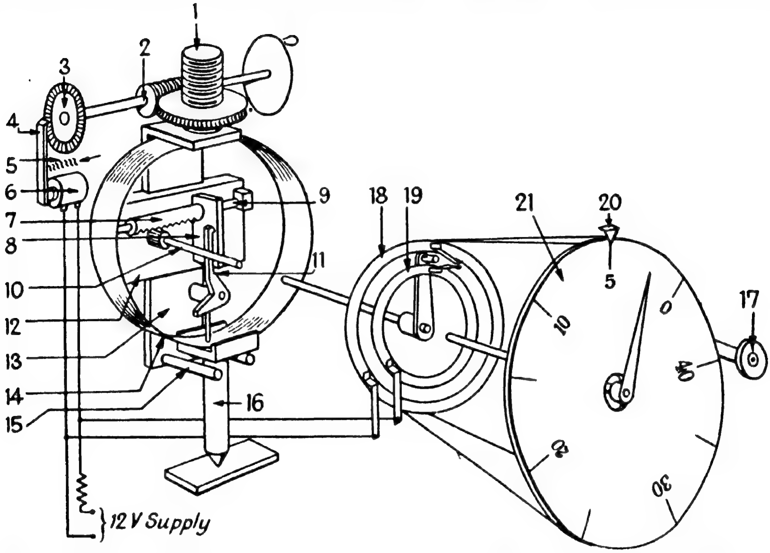


FIG. 69. FIRTH-BROWN VARIABLE LOAD HARDOMETER—ARRANGEMENT OF LOAD MECHANISM

1. Load Screw.
2. Worm Shaft.
3. Ratchet Wheel.
4. Pawl and Armature.
5. Armature Spring.
6. Magnet Coil.
7. Rack
8. Rack Plate.
9. Rack Guide.
10. Indicator Spindle.
11. Bell Crank Lever.
12. Fork Guide.
13. Fork.
14. Load Spring.
15. Limit Pin.
16. Indenter Spindle.
17. Load Adjuster.
18., 19. Collector Rings with Contacts.
20. Load Pointer.
21. Load Dial.

refinement is the reading of the diameter or diagonal of impression by means of a counter outside the eyepiece. This is a decided advantage over scale reading as employed on the standard type of tester.

A diagrammatic arrangement is illustrated in Fig. 69. The loading control mechanism is of ingenious design, and consists of an accurately ground compression ring, the smallest deflection of which is transmitted by means of a micrometer device to a rotating pointer, which in turn trips an electrical relay. In this way the motion of the loading handwheel is arrested as soon as the predetermined load is attained.

Referring to Fig. 69, the load with which the test is to be made is selected by turning the adjusting knob (17) until the load required comes

under the load pointer (20). This action carries the electrical contact mounted on the slip rings (18) and (19), through a similar angular movement away from the normal resting position of the mechanism when the spring is at zero compression.

To operate the mechanism, the component to be tested is placed on the anvil under the indenter (16) and the handwheel revolved clockwise. Then, through the medium of the worm, worm wheel and the loading screw (1), the fork (13) together with the load spring (14), the indenter (16) and the bell-crank lever (11), will move downwards with no compression on the spring until the indenter (16) meets the testpiece. During the downward movement the upper end of the bell-crank lever moves down the face of the rack plate (8), and this, being parallel to the fork (13), experiences no horizontal movement along the rack guide (9). There is thus no movement of the indicating pointer on the load dial (21) during this preliminary advance.

The continued loading carries the fork (13) downwards, leaving stationary the lower end of the load spring (14) and also the actuating pin between the spring and the bell-crank lever (11). This motion, which is accompanied by compression of the load spring, causes a partial revolution of the bell-crank lever, which pushes the rack plate (8) and the rack (7) to the right along the rack guide (9). This action causes rotation of the spring controlled indicator spindle (10), and the contact rotating between the slip rings will short-circuit the spring contacts carried on these rings. When this occurs the dial indicating pointer will have moved to the zero mark on the dial.

The establishment of the electrical contact short-circuits the magnet coil (6) and allows the spring (5) to push the pawl (4) into a tooth of the ratchet wheel (3), thus arresting the worm shaft (2) and preventing any further loading.

HARDNESS TESTING IN GENERAL

There are many types of hardness-testing machines available. Some work on the Brinell or Diamond principle, the measurement of the impression being one from which the indentation area is calculated. Others, e.g. the Rockwell and the Avery, are direct-reading, as indications of hardness are given on a dial according to depth of indentation which is measured.

For general industry the type of hardness tester chosen is governed solely by individual applications or requirements. Some machines are more adapted for rapid testing of uniformity of material rather than absolute hardness values, while others predominate for definite hardness indication.

While, therefore, the machine which suits the job can be chosen for industrial requirements, for standard testing to British Specification requirements only certain machines may be used. These must conform to either B.S.S. No. 240 or 427, which deal with Brinell and Diamond Pyramid machines respectively. Reference to these specifications will denote that hardness values must be based upon area of indentation calculated from the mean diameter or diagonal, *and not from the depth of the impression.* This definition then immediately bars direct-reading

hardness machines for standard testing, as they denote hardness according to depth. Also the accuracy of measurement of the indentation is specified. For Brinell-type machines this for a 10 mm. ball impression is $\pm$ 0·025 mm., for a 5 mm. ball $\pm$ 0·01, and for a 1 mm. ball $\pm$ 0·002 mm. For the Diamond Pyramid type the accuracy of measurement of the diagonal of indentation must be $\pm$ 0·001 mm. These requirements must be borne in mind when choosing equipment for British Standard testing.*

NOTCHED-BAR IMPACT TESTING

In order to clearly understand the reason for impact testing, reference must be made to a quotation from the work of a well-known investigator. Stanton in 1921 said: "The conclusion seems unavoidable that the value of the impact test lies, not in discriminating between the impact resistance of different materials, but as a means of ensuring that the impact strength of any given material is at its highest." Also: "The chief value of the test will be its use as a means of detecting a dangerous condition of microstructure due to faulty heat treatment."

The above statements are of fundamental importance if a clear conception of impact testing is to be held. They clearly indicate that the test is not primarily concerned with the shock-resisting properties of materials. The shock loading is only the means of test, and the results obtained do not indicate that a material having a high impact value is better able to resist shock than one having a much lower value. In point of fact, slow bending tests on notched bars give comparable results with the impact type of test.

The value of the notched-bar impact test, then, is to prove that any given material has a certain impact value which should be obtainable if that particular material has been correctly treated.

The importance of the test is that material which has been correctly treated can be discriminated from the same material incorrectly treated, which condition cannot be determined by the ordinary tensile test results. Tensile test results do not ascertain temper brittleness in nickel-chrome steels, but an impact test will quickly do so. In some cases a low impact value does indicate a tendency to fracture under shock loading, but too liberal an interpretation of this will not hold true

It is for the above-mentioned reasons that notched-bar impact values are only comparable within the same class of material, i.e. the value for a mild steel has no relation to that for a nickel-chrome steel. Again, one material may be more notch-sensitive than another, and whereas two unnotched specimens might require the same energy to fracture, when notched the two materials would give widely differing values. From this it would appear that the Izod value is not significant until a notch is made at which stress concentration can occur.

The direction in which the notch is cut relative to the grain flow also has a marked effect on the impact value obtained from the same material. This direction is indicated subsequently.

* Copies of British Standard Specifications Nos. 240 and 427 are obtainable from the British Standards Institution, 28 Victoria Street, London, S.W.1, price 2s. 3d. each post free.

The Izod Impact Test. This is the accepted British Standard impact test called for in specifications. It originated through a nicked-bar fracture test carried out by Izod in 1903 when investigating defective material from a burst gun barrel. He found that, whereas all other known tests would not discriminate between the good and the defective material, hand-notched bars fixed in a vice and struck with a hammer immediately did so. The defective material broke in a brittle fashion with one blow, whereas the good material took several blows to fracture.

It must be remembered that no form of impact test was in use at the period, and Izod used his discovery as the basis of the pendulum impact machine.

The Izod Machine. The machine (Fig. 70) consists of a base casting upon which are erected two rigid A-frames. Between the apex of these frames is swung a pendulum, mounted in ball bearings in such a way that all lateral movement is prevented. The pendulum weight is fitted with a plug into which a hardened steel insert is dovetailed which strikes the specimen at a fixed distance above the surface of the gripping dies, which are held in a vice secured to the base casting.

A loose pointer actuated by the pendulum moves over a quadrant indicating, on the left-hand side facing the machine, the impact value in foot-pounds. Some machines have the scale above the pendulum standard, and in this case the energy to cause fracture is read off on the right-hand side facing the scale, i.e. the initial energy of the pendulum before impact minus the residual energy after impact. The striking velocity of the pendulum conforms to British Standard requirements, i.e. not less than 3 metres per second.

The specimen is positioned in the vice by a gauge, to ensure that the point of impact is at standard distance from the notch, the notch facing the direction of the blow.

Specimens. Specimens used may be either one- or three-notch 10 mm. × 10 mm. or 0·45 in. diameter section as shown. When the three-notch specimen is used, the second notch is cut to be at right angles to the first and third notches. (See Figs. 72A and B.) It should be noted that the notches must be cut at right angles to the direction of rolling for the standard test. This point is mentioned because, although for many bars, etc., the test piece can only be prepared with longitudinal grain flow, with large bars or forgings the single-notch specimen could be cut in a number of ways, all giving different impact values. Subsidiary standard specimens of 10 mm. × 5 mm. or 5 mm. × 5 mm. cross-section can be used where there is not sufficient material to cut the standard specimens, but unfortunately the impact values obtained from these have no constant proportion with the values from the full-size specimens.

As shown in the diagram, the notch angle is 45° for all specimens, and the root radius is 0·25 mm. With subsidiary standard test pieces the depth of notch is 1·5 mm.

Although brittleness under impact is more marked the smaller the notch angle and root radius, the above values have been standardized from the results of investigations, and in the case of the root radius because a dead-sharp notch cannot be easily produced.

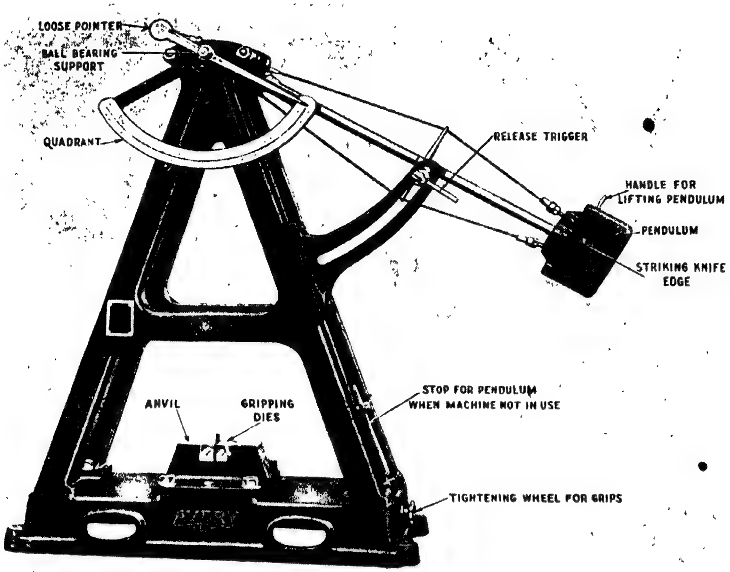


FIG. 70. IZOD IMPACT TESTER

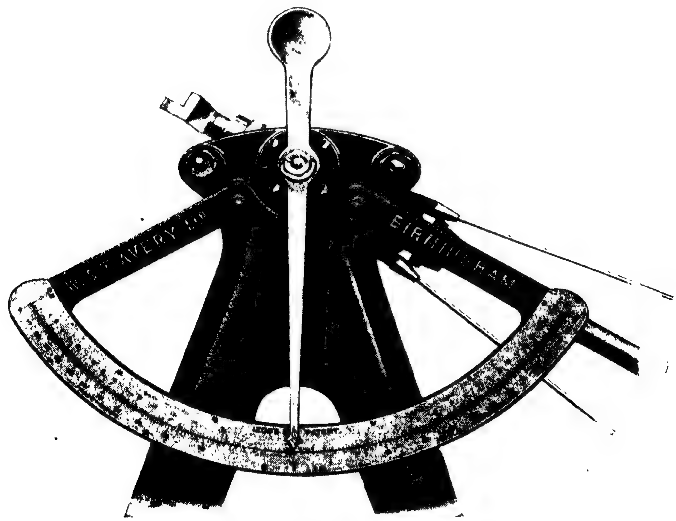


FIG. 71. SCALE ON IZOD MACHINE

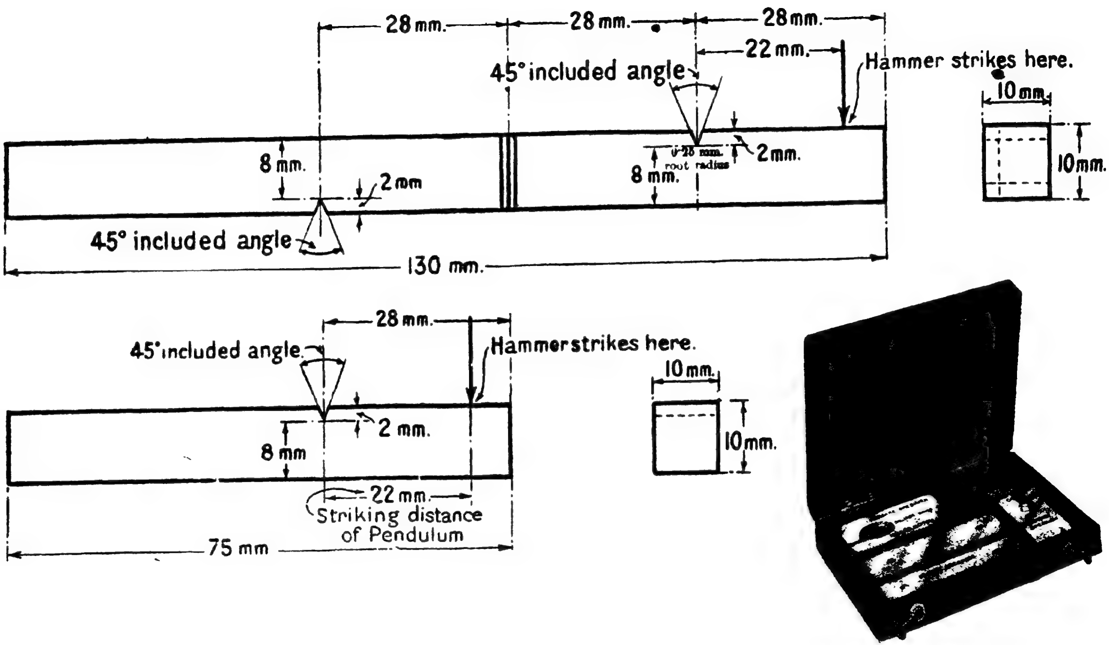


FIG. 72A. IZOD STANDARD TEST PIECES AND GAUGES

Gauges for Specimens. A set of gauges consisting of a notch gauge, calliper gauge, and a positioning gauge are supplied for both the square- and the round-type specimens. These are seen in Fig. 72A. A gauge having

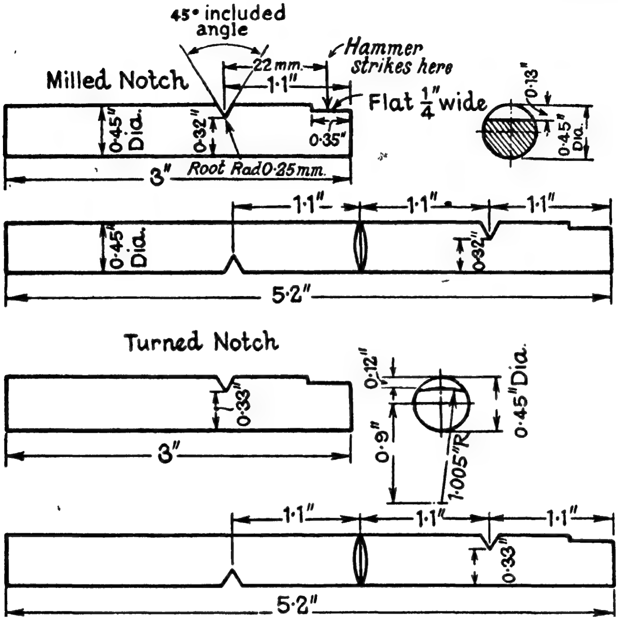


FIG. 72B ROUND IZOD TEST PIECES
The flat on the specimen is generally omitted

a standard notch cut in it is also provided. The notch gauge is for checking the profile of the notch, the calliper gauge for checking the depth of metal behind the notch, and the positioning gauge is for setting up the specimen in the vice.

(N.B. Three-notch Izod test specimens must be used when testing to A.I.D. requirements.)

CHAPTER VI

THE CALIBRATION OF TESTING MACHINES

In order that testing machines may remain accurate in their indications it is essential that they be periodically calibrated.

On most machines routine checks can be carried out to ensure that accuracy is being maintained, but for large tensile-testing machines only the makers have the necessary equipment to effect a complete calibration. The initial calibration of a testing machine is naturally the concern of the manufacturer, but subsequent calibration is the responsibility of the user, and firms approved by the A.I.D. for testing materials must ensure that this calibration is effected at defined intervals.

Tensile-testing Machine. The Board of Trade will supervise such a calibration and issue a certificate if results are satisfactory. Alternatively, the makers of the machine undertake the calibration. It is not intended to cover in detail the many points concerned in checking tensile machines, but to give main outlines in order that the principles may be understood.

For single-lever machines, employing a single counterpoise and two knife-edges, dead loads are the most satisfactory method of effecting complete calibration. In fact, whenever possible, dead loads should be used, and these loads must be certified by H.M. Weights and Measures. The various factors in the calibration of a single-lever machine are listed below—

1. Weight of counterpoise.
2. Condition of knife-edges and their seatings. The load per inch of effective bearing length of knife-edge must not exceed 5 tons, i.e. capacity in tons divided by bearing length of knife-edge must not exceed 5.
3. Measurement of distance between knife-edges.
4. Accuracy of marking of load scale.
5. Balance of lever. This is checked by placing a known load on the hanging shackle, and with a much smaller weight placed at such a distance from the lever fulcrum as to exert an equal and opposite turning moment. The lever will first be balanced by the counterpoise for this operation, the top specimen holder being in position.

Fig. 73 will show the method. The hanging shackle is loaded to 10 cwt. The moment about the lever fulcrum will be 1120 × 2 lb. in. Then for the balance of the lever to be restored, 56 lb. must be at such a distance from the fulcrum that—

$$1120 \times 2 = X \times 56$$
$$\therefore \quad X = 40 \text{ in.}$$

The 56 lb. weight is hung on at this distance, and the lever should again be in balance. 56 lb. can then be removed and the counterpoise moved along to balance the 10 cwt. The distance moved should be, assuming the counterpoise to be 10 cwt.—

$$1120 \times 2 = 1120 \times L$$
$$\therefore \quad L = 2 \text{ in.}$$

6. Sensitivity. With the beam balanced as above, the least additional weight which when placed on the hanging shackle causes the beam to deflect from its balanced position is a measure of the sensitivity. For a 20-ton machine it should be about $1\frac{1}{2}$–2 lb.

7. The machine must be calibrated in the position in which it is to be used.

Calibrating Levers. For machines of large capacity it is not possible to load sufficient weights on to the hanging shackle, so that a lever arrangement is fitted between the testing grips or compression plates, and by means of proportional weights the load exerted can be accurately measured. The counterpoise is then moved along until the lever of the testing machine proper is in balance, and such movement when multiplied by the counterpoise weight should equal the product of the dead load

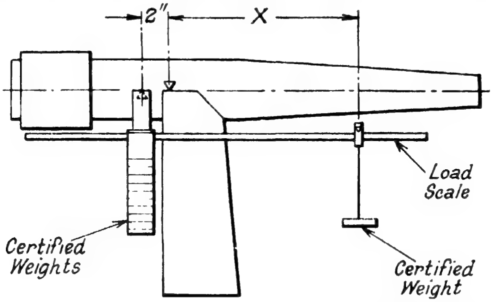


FIG. 73. DIAGRAMMATIC ILLUSTRATION OF CHECKING SINGLE-LEVER MACHINE FOR BALANCE

and knife-edge distance in the same manner as the previous example, i.e. the counterpoise vernier should indicate the correct dead load on the load scale.

Such levers are termed *calibrating* or *proving* levers, and an illustration of calibration by such means by W. & T. Avery Ltd. is given in Fig. 74. The lever system is shown diagrammatically in Fig. 75.

Proving Ring. Alloy steel rings fitted with a micrometer measuring device across the internal diameter form another means of calibration of a testing machine. They can be made for checking a machine in both tension and compression, and an illustration of the ring fitted in readiness for a tensile calibration is given in Fig. 76.

The rings which are made for various capacities are initially very accurately calibrated under dead loads, the micrometer readings of the deflections which are taken forming the basis for the calibration of other machines.

The subsequent method employed is to load the ring by increments in the machine to be calibrated and hold the load steady at some particular value as indicated by the machine. The micrometer reading then determines the actual true load on the ring, with which the load indication of the machine must agree within the permissible tolerance.

Machines have usually a certified accuracy of 0·5 per cent throughout

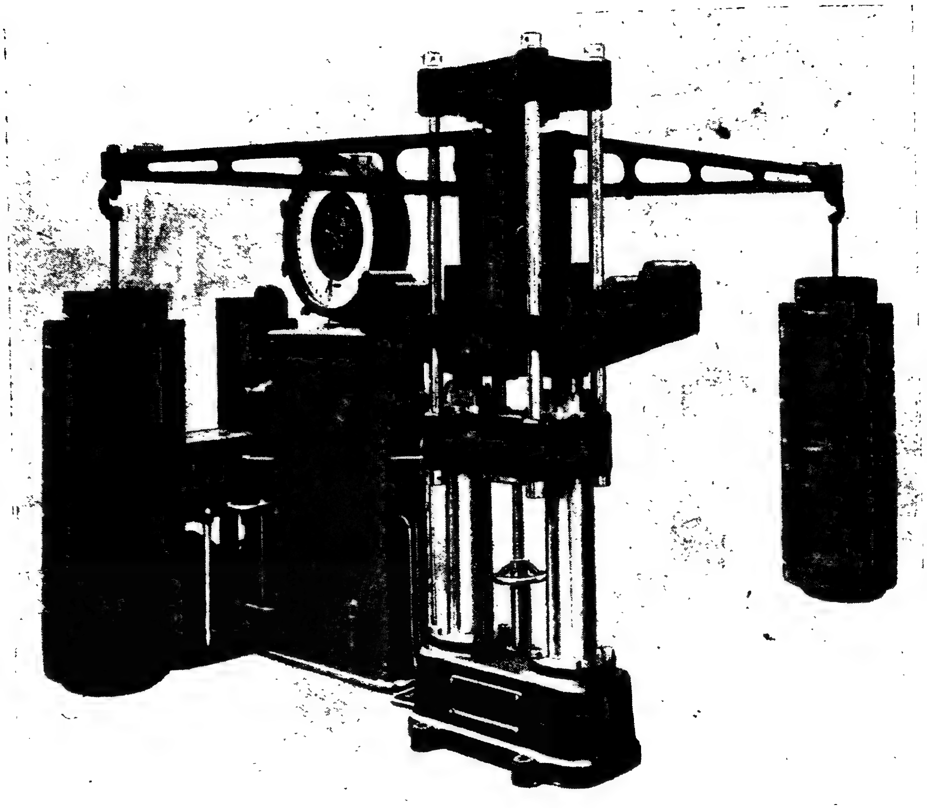


Fig. 74. Calibration by Proving Levers of Avery Universal Machine

their range, i.e. 11·2 lb. per ton, but the actual accuracy of the modern machine is usually much finer.

Standardizing Boxes. These so-called "boxes" are hollow test bars filled with mercury, and loads are measured by the increase in volume of the hollow bar. This is accomplished by providing a horizontal capillary on one side, and a plunger actuated by a micrometer screw on the other side of the test bar, both communicating with the mercury-filled interior.

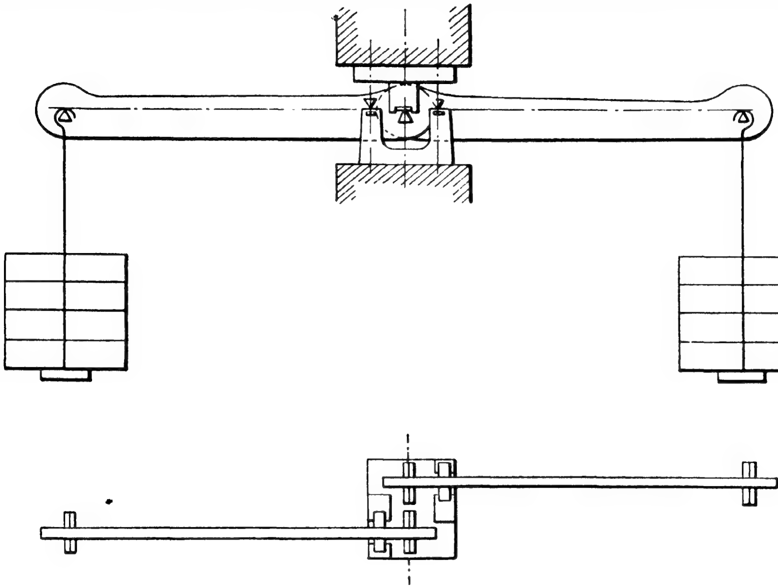


FIG. 75. DIAGRAMMATIC ARRANGEMENT OF PROVING LEVERS

The capillary is also partly filled with mercury and has a zero marking. When fitted between the tension grips of a tensile machine and subjected to load, the increase in volume alters the position of the mercury. This is then returned to the zero mark by means of the micrometer screw.

The standardizing box is itself calibrated by dead loads, and micrometer readings against load recorded. These readings form the basis for measuring the actual load when the box is used in a tensile machine. It is seen that the box takes the place of calibrating levers, and being very compact and quickly set up forms one of the simplest methods of checking. The accuracy is $\pm$ 0·5 per cent for loads above one-tenth of the maximum capacity of the particular box used, i.e. after a tensioning stress has been first applied. They are made by A. J. Amsler & Co. and marketed by T. C. Howden & Co. of Birmingham.

Standard Test Bar. In this method a test bar is made from material whose stress/strain characteristics have been accurately determined, and whose line of proportionality is straight within the limits of elastic extension.

The test bar is accurately made and is of such dimensions that maximum capacity of the testing machine does not exceed 75 per cent of the limit of proportionality of the material.

Then, using the test bar in the machine to be tested, an A.I.D. approved-type extensometer (e.g. the Lindley) is fitted, and readings of load and extension are taken at various increasing loads in the manner

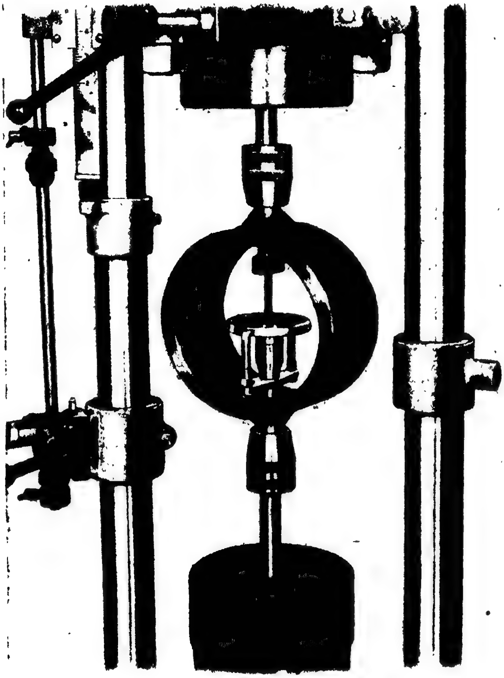


FIG. 76. PROVING RING FOR CALIBRATING IN TENSION
(Avery)

of a tensile test. These readings when plotted should be on a straight line coinciding with that previously determined for the material. This test is more in the nature of a routine check between periods of complete calibration.

Hardness-testing Machines. Standard hardness specimens, supplied by the manufacturer of the particular testing machine, form the means of checking the accuracy of hardness-testing equipment. By using standard specimens of different hardness, the full range of the machine can be checked.

In addition, the focus of measuring microscopes using graticule scales must be checked, as with some microscopes, if focus is obtained by

alteration of the wrong tube, the scale ceases to be correct. A standard scale can be used for checking the microscope.

A small lever system similar in construction to that illustrated in Fig. 75, can be used for checking the actual dead loads applied to the indenter.

Izod Impact-testing Machines. Initially, the impact machine is calibrated by direct measurement of the weights, angles, and dimensions involved. The centre of percussion* should coincide with or be slightly above the striking edge in the tup and by using the formula derived from a simple pendulum it is an easy matter to check whether the measured distance between the centre of rotation and the striking edge is the same as that between the centre of rotation and the centre of percussion.

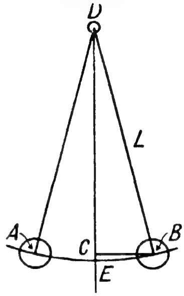


FIG. 77. SIMPLE PENDULUM

Referring to Fig. 77, if the weight of the suspending arm L is neglected and the pendulum is allowed to swing so that the distance BC is small in comparison with L, then DE may be assumed to be equal to DB, i.e. L, and the time of one complete swing is given by the expression

$$T = 2\pi\sqrt{\frac{L}{g}}$$

where T = time in secs. of one complete swing (i.e. from B to A and back again to B)

L = length in ft. of pendulum from the centre of rotation to the centre of percussion.

$g = 32{\cdot}2$

Solving for L,
$$T^2 = 4\pi^2\frac{L}{g}$$

$$L = \frac{g}{4\pi^2} . T^2$$

$$= 0{\cdot}815T^2$$

When making the check, allow the pendulum to swing over about 10 deg. and count the number of complete swings in one minute. The time T for one swing is then 60/no. of swings.

For the actual Izod pendulum, and counting the number of complete swings only, the distance in feet of the centre of rotation from the centre of percussion is given more closely by

$$L = 0{\cdot}81T^2$$

The result of this calculation should then closely agree with the measured distance between the centre of rotation and the striking edge.

To check the potential energy the weight of the pendulum must be determined and this may be done by an arrangement such as illustrated in Fig. 78

When the pendulum is elevated on to its normal trigger stop, the weight multiplied by the height H should equal 120 ft. lb. $\pm$ 2 per cent for the

* If a body oscillates about an axis then the centre of percussion is that point at which if a blow is struck by the body the percussion action is the same as if the whole weight of the body were concentrated at that point.

standard machine. If the machine has been correctly installed, subsequent periodical checks of the following points will ensure continued accuracy—

1. With a level on the top surface of the vice, see that the machine is vertical on its foundations.

2. Operate the machine normally but without a specimen. The loose pointer should then indicate zero.

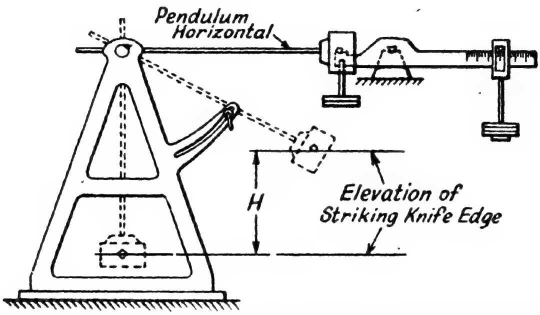


FIG. 78. ARRANGEMENT FOR CALIBRATION OF IZOD MACHINE

3. Hang the pendulum vertically. The loose pointer should register the central zero.

4. With the pendulum as in (3) the striking knife-edge should just be in contact at a height of 22 mm. above the gripping dies with a standard specimen fixed in the vice.

5. That the striker knife-edge and the edge of the fixed jaw of the vice are in good condition.

[*Note.* Knife-edges for tensile and impact machines are made from iron-cored steel in order that the shocks transmitted to the knife-edge should not cause fracture. The iron core prevents cracking of the hardened knife-edge itself.]

CHAPTER VII

TESTS ON SHEET, STRIP, WIRE, AND TUBE

THE test piece shown in Fig. 79 is used for metal sheet and strip material not exceeding 0·128 in. (10 S.W.G.) in thickness, and is suitable for both steel and duralumin. For aluminium, the test piece should be made as indicated at (*b*). These two types of test pieces cover the requirements of most specifications.

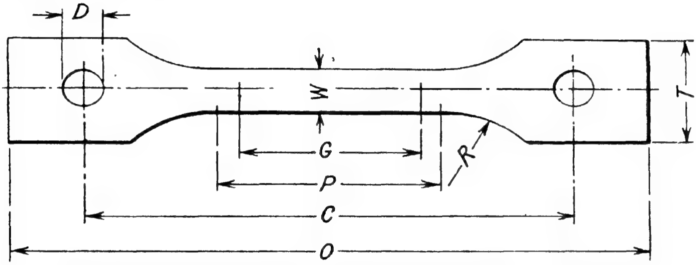


FIG. 79. TEST PIECE FOR THIN METAL SHEET AND STRIP

	Parallel length P	Gauge length G	Width W	Radius R	Locating centres C	Diameter D	Overall Length O	Overall Width T
	in.	in.	in.	in.	in.	in.	in.	in.
(*a*)	$2\frac{1}{2}$	2	$\frac{1}{2}$	1	6	$\frac{5}{16}$-$\frac{1}{2}$	8	$1\frac{1}{4}$
(*b*)	$2\frac{1}{2}$	2	$\frac{3}{4}$	$3\frac{1}{2}$	$7\frac{1}{2}$	$\frac{5}{16}$-$\frac{1}{2}$	$9\frac{1}{2}$	$1\frac{1}{4}$

Percentage elongation on sheet and strip aircraft material is only specified on material of 12 gauge (0·104 in.) and over, so that it is best to adhere to one particular width of test piece when dealing with such gauges in duralumin, as the gauge length is 2 in. for all gauges.

Unless the thickness is constant there is no relationship between gauge length and cross-sectional area of the rectangular test piece that will give the same percentage elongation, as with round test bars.

If the gauge of the material is constant and only the width varies, the percentage elongation is approximately constant if the following is observed—

$$\text{Gauge length in.} = C \times \sqrt{(\text{cross-sectional area sq. in.})}$$

where C is a suitable number which will provide a convenient gauge length.

For testing aircraft material the grips of the testing machine must be provided with dowels which accurately locate the centre-line of the test piece with the line of pull. The test piece must therefore be symmetrical about the centre-line passing through these locating holes, and to ensure that this is so, specimens must be prepared by means of a jig. Test specimens must be accurately measured before testing, as small variations in gauge will lead to substantial errors in the results obtained. A piece may be sent as 16 gauge, but instead of measuring 0·064 in. would more

likely measure 0·068 in., due to tolerance allowed on the nominal thickness. The following example shows the importance of gauge measurement when dealing with specimens of small cross-sectional area—

1. Cross-section of test piece = 0·028 in. (gauge) × 0·500 in.
   Area = 0·014 sq. in.
2. If measured as 0·029 in. × 0·500 in.
   Area = 0·0145 sq. in.

Maximum load = 0·382 ton.
Maximum stress as (1) = 0·382/0·014
= 27·29 tons/sq. in.
Maximum stress as (2) = 0·382/0·0145
= 26·34 tons/sq. in.

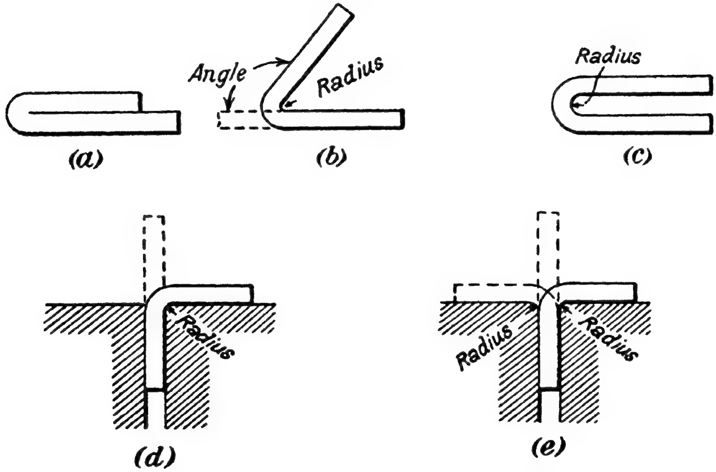


FIG. 80. BEND TESTS ON SHEET AND STRIP MATERIAL

It is also usual to specify the direction of rolling of a sheet in relation to the longitudinal axis of the test piece cut from it. For sheet and wide strip, the direction of rolling must be at right angles to the longitudinal axis of the specimen.

Bend Tests. Numerous specifications for sheet and strip material include bend tests, to be carried out on material 12 S.W.G. (0·104 in.) and thinner. Results obtained from such tests give more information about the ductility than the percentage elongation. The tests are carried out on strips ½ in. wide, cut so that the longitudinal axis of the test piece is at right angles to the direction of rolling, except in the case of strip material, when the longitudinal axis has necessarily to be parallel with the direction of rolling.

In order that failure under test should not occur due to cracks being initiated by sharp corners, the longer edges of the test piece must be smoothed and chamfered so that the cross-section has semicircular ends.

The length of the piece is governed simply by the particular method of gripping and bending employed. The actual tests are of two kinds, (1) Single-bend, (2) Reverse-bend.

*Single-bend Tests.** These are carried out in one of three ways—

(*a*) Bending flat;

(*b*) Bending through an angle over a specified radius;

(*c*) Bending through 180° over a specified radius.

Examples of each case are given in Fig. 80. The radius over which the bend has to be made is given in terms of the nominal thickness of the material or else direct, and the test piece has to withstand such bending without cracking on the convex surface.

*Reverse-bend Tests.** These are carried out by repeated bending over a specified radius, through either 90° or 180°. In either case the initial bend through 90° is not counted. The bending of the test sample is continued for a specified number of reversals or until the test piece breaks. In addition to individual test pieces having to satisfy the requirements of the above tests it is often necessary to carry out a single-bend test as (*c*) on the corner of every sheet.

Bend tests can be carried out with simple apparatus, and a suitable machine for light gauges is illustrated in Fig. 81. It is also possible to make the bend test as (*b*) or (*c*) by pressing the specimen into a die by means of a press tool having the specified radius.

For steel sheet and strip it may be carried out by pressing the specimen into lead by means of an appropriate former. This has the advantage of making a perfect die regardless of slight variations in gauge due to manufacturing tolerance.

Tests on High-tensile Steel Wire. High-tensile steel wire is obtained by a cold-drawing process. This greatly increases the tensile strength, but with a corresponding reduction in ductility. In order to ensure that the required tensile strength is obtained without undue brittleness, the following tests are carried out: (1) Tensile, (2) Torsion, (3) Bending, and (4) Wrapping.

Tensile Test. This is carried out on a sample length of the wire, and must give a minimum stress figure in accordance with the requirements of its particular specification. As the rate of loading influences results obtained on hard-drawn wire, reference to the appropriate British Standard Specification should be made if standard tests are to be undertaken.

Torsion Test. This is a ductility test, and will also show any longitudinal defects in the wire due to the cold-drawing operation. The wire test sample is gripped in two holders so that the length to be tested is $100d$, where d is the diameter of the wire. One holder is fixed, and the other which is rotatable is driven by hand power, thereby twisting the wire. The number of revolutions to cause fracture is denoted by an indicating device on the machine.

The specification usually calls for a certain minimum number of turns before fracture, on a basic length of $100d$. A machine suitable for this

* This general information on single-bend and reverse-bend tests, including Fig. 80, is abstracted by permission from British Standard Specification No. 485, official copies of which can be obtained from the British Standards Institution, 28 Victoria Street, London, S.W.1, price 2s. 3d. post free.

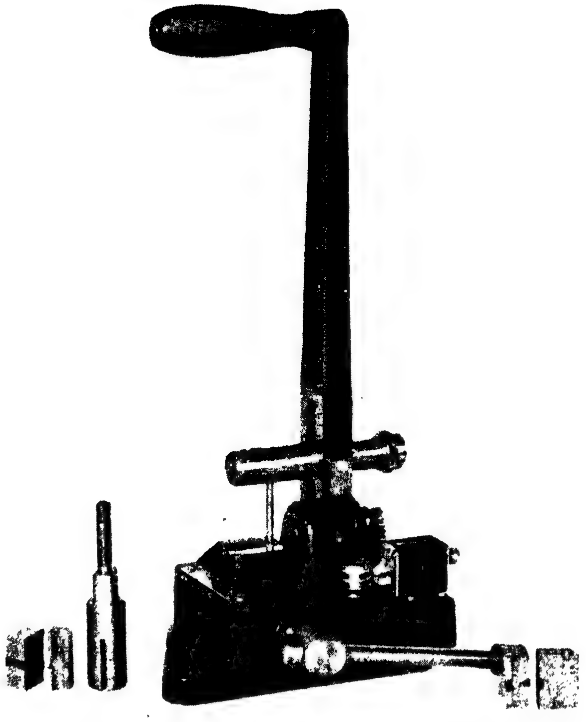


FIG. 81. AVERY MACHINE FOR WIRE AND THIN SHEET TESTS

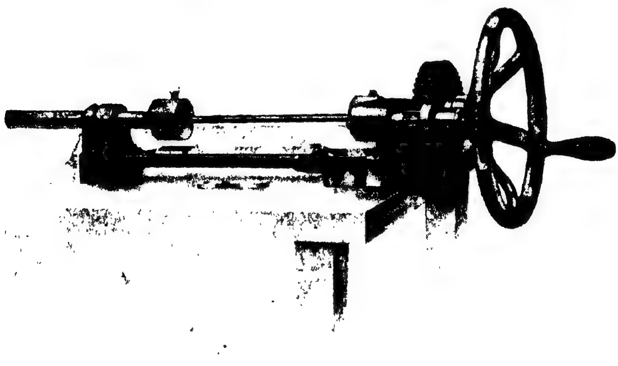


FIG. 82. AVERY TORSION MACHINE

test is illustrated in Fig. 82. It is also capable of carrying out alternate twisting through 180°.

Bending Test. This is similar to the 180° bend test specified for thin metal sheet and strip, and is carried out in the type of machine shown in Fig. 81. As with the sheet or strip test piece, the number of reversals to fracture is noted, the first through 90° not being counted.

Wrapping Test. For this, the end of each coil or hank of the wire must withstand, without showing any fracture, being wrapped round its own diameter eight times, and unwound with the exception of the last turn. Remembering that the wire has been cold drawn to a tensile strength of about 80–100 tons/sq. in., this test is very severe, and any tendency to brittleness will cause fracture.

Tests on Steel Tube. For maximum stress and proof stress determinations, sample lengths of the tube can be tested whole, in which case the ends are either plugged or flattened for gripping in the testing machine. Alternatively, a test sample can be cut from the wall of the tube. If the latter method is employed, the specimen is similar to that indicated in Fig. 79. In this case, however, the parallel length of $2\frac{1}{2}$ in. must not be flattened, and the half-inch width is the chordal width of the curved surface. The ends of the test piece can be flattened for gripping in the testing machine.

For steel tube a 0·2 per cent proof stress is usually specified.

Proof Bend Test.* This consists of applying a specified bending moment to the tube, which is supported at two points and loaded at a third. The elastic deflection due to this bending moment is calculated for a point close to the application of the load, and the permanent set due to the moment measured at the same point. For the tube to pass the test, the measured set must not exceed X per cent of the calculated elastic deflection. X is specified in the particular specification relating to the tube.

Flattening Test.* These tests are carried out on a sample piece cut from a tube. The short length so cut is then flattened between flat faces by hammer blows. The degree of flattening is specified for each type and specification of tube.

* Abstracted by permission from British Standard Specifications T35 and T45, official copies of which can be obtained from the British Standards Institution, 28 Victoria Street, London, S.W.1, price 1s. 3d. each post free.

Specification T45 contains a description and illustration of an approved type of testing machine for carrying out the proof bend test.

CHAPTER VIII

RADIOLOGICAL TESTING AND CRACK DETECTION

Nature of X-rays. The well-known spectrum of visible light, the colours from red to violet dispersed according to wavelength, is actually but a small section of the complete spectrum of all electro-magnetic radiations, which comprise radio waves, heat waves (infra-red), visible light, ultra-violet light, X-rays, γ-rays from radium, and cosmic rays. These radiations are of a similar nature and differ only in wavelength, radio waves being the longest and cosmic rays the shortest. The wavelength of X-rays is of the order 10^{-8} cm.

Properties of X-rays. The outstanding property of X-rays, and one which accounts for their main application in industry and medicine, is that of being able to penetrate solid matter. All materials are partly transparent to X-rays, to a degree which depends on the elements contained in the material and on the wavelength of the rays. In passing through matter the beam becomes attenuated, owing to the action of the atoms which scatter the rays in all directions and also absorb energy when they are excited and made to emit secondary X-rays. This loss of intensity is termed absorption, and it follows the relation—

$$I = I_0 e^{-\mu t}$$

where I_0 = initial intensity,

I = intensity after traversing a thickness t of the material,

μ = constant,

e = exponential function.

For a given wavelength this constant μ depends on the concentration of electrons in the absorbing material, and is therefore a function of the atomic number, or approximately the atomic weight, and of the number of atoms in the path of the rays. The result is that absorbing power is roughly dependent on the density. Lead and dense metals absorb strongly, hence the extensive use of lead in practice to destroy unwanted radiation. Copper absorbs more strongly than iron, iron than aluminium, and aluminium than magnesium. Materials of organic origin, such as wood, contain light elements, e.g. hydrogen, oxygen, and carbon—and are therefore easily penetrated. μ also depends on the wavelength in such a way that shorter wavelengths suffer less absorption than longer waves, i.e. have greater penetrating power.

Other effects of X-rays are photographic action on sensitized films, excitation of fluorescence in certain salts (both effects being the normal means of utilizing the rays), various biological effects, etc.

Production of X-rays. The X-ray tube is one of many devices, like the cathode-ray tube, the radio valve, etc., which depend on the passing of electrons from one electrode to another in vacuo. In all these the electron source is an incandescent electrode—usually a filament heated

by the passage of an electric current. A body in such a state emits electrons which can be made to perform various functions, according to the purpose of the apparatus, by means of electric and magnetic fields and suitable electrodes. In the X-ray tube, the electrons emitted by the filament or cathode are accelerated by means of a strong electric field resulting from a high tension of many thousand volts between the cathode and anode. When electrons possessing a high velocity, i.e. great kinetic energy, strike a material, X-rays are generated by virtue of their sudden stoppage. Thus to generate X-rays all that is required is to put an anode in the path of the fast-moving electrons. This anode is generally termed the *target* of the tube, since it is continuously bombarded and made to radiate X-rays.

These rays have a range of wavelengths, the spectrum of the radiation being like the visible continuous spectrum of an incandescent body. The distribution of energy in the spectrum follows a curve of the form shown in Fig. 83, having a maximum and definite limit on the short-wave side. For practical purposes it is only necessary to consider a band of wavelengths on each side of the maximum, since the longer and less penetrating waves are mostly absorbed in the tube window. The wavelength of maximum intensity decreases with increasing electron energy or with increasing voltage, so that there is a definite relation between penetrating power and applied voltage.

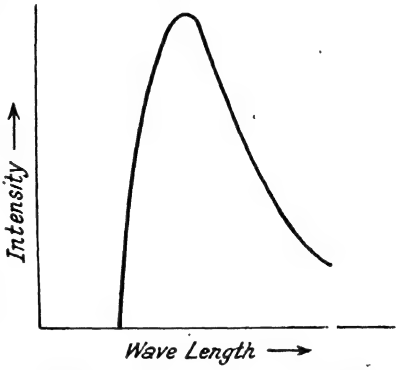


FIG. 83. DISTRIBUTION OF ENERGY IN THE X-RAY SPECTRUM

In actual practice the electrons are made to converge on to a focal spot on the surface of the target so that the rays originate approximately at a point—an essential for the formation of images on the fluorescent screen and the radiograph.

From the point of view of energy an X-ray tube has a low efficiency. Only about 2 per cent of the input energy is transformed into X-rays, the remaining percentage being converted into waste heat, hence the need for efficient cooling is of paramount importance. Moreover, since this heat is developed at the focal spot on the surface of the target, it must be dissipated very rapidly if the target is to remain intact. A tube run at 100 kV. and 10 mA. will generate approximately a kilowatt of energy as heat. This is equivalent to a small electric fire concentrated at the focal spot. Air cooling is employed in the smaller tubes, while larger tubes are water- or oil-cooled.

Further requirements of X-ray tubes are (*a*) that they should be rayproof, i.e. emitting rays only through the prescribed window, and (*b*) that they be made shockproof by enclosing in an earthed shield.

X-ray Installations. An X-ray installation consists essentially of the tube, a high-tension transformer with a tapped auto-transformer in the

primary circuit as a means of varying the voltage applied to the tube, and a filament transformer in the high-tension circuit. A rheostat in the primary of the latter controls the filament temperature, and hence the electronic emission, which in turn determines the discharge current (a few milliamperes) from the cathode to the target. Small X-ray outfits have alternating current applied to the tube, as the latter is self-rectifying, i.e. the alternating current is rectified to direct current by the X-ray tube itself. More powerful installations are supplied with high-tension direct

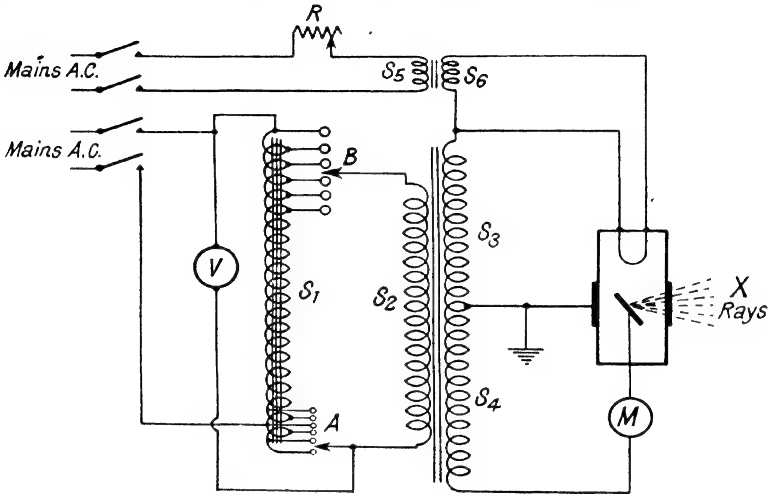


FIG. 84. SIMPLIFIED SCHEMATIC OF X-RAY OUTFIT (SELF-RECTIFYING X-RAY TUBE)

A. Mains voltage compensator.
B. Kilovoltage selector.
S_1. Auto-transformer.
S_2. High-tension transformer primary.
S_3 S_4. High-tension transformer secondary.
S_5. Filament transformer primary.
S_6. Filament transformer secondary.
R. Filament regulator.
V. Voltmeter indicating compensated voltage.
M. Milliammeter (for anode current).

current from a separate circuit containing one or more rectifying valves and condensers. A typical circuit is shown in Fig. 84. The two variables in the operation of such a plant are the high-tension selector and the filament control. The latter affects only the quantity of X-rays generated and has no effect on the penetrating power, which is determined by the applied high tension. The higher the voltage, the greater will be the penetrating power. The selection of a suitable high tension is therefore conditioned by the nature of the object under examination.

The Technique of Radiological Testing. The object to be examined, say a casting, is placed between the source of X-rays and a fluorescent screen or photographic film (suitably protected from light by black paper or a special cassette) and as close to the latter as possible. The object casts its X-ray shadow, as it were, on the screen or film, but this shadow is not a mere silhouette: the area within the shadow shows patches of

light of varying intensity, depending on how much absorption each ray has suffered in traversing the object. The amount of absorption depends on the nature of the material and on the thickness in the path of the rays, thus causing the image to disclose all variations in these factors within the object. Flaws and cavities can be seen because such unsoundness means lack of absorbing matter, and therefore a cavity shows up on the screen as a bright spot or on the film as a black spot.

The fluorescent screen method is rapid, but not so sensitive as the slower photographic method. Fine detail can be shown on the film, and it is possible to use lower voltages than with screening. A high tension can be used in photography which would not produce sufficient illumination on the fluorescent screen, because in the former instance the energy is built up in the sensitive emulsion during the exposure (usually a matter of minutes). From a consideration of the law governing absorption it is easy to show that reducing the high tension increases the contrast, and hence enhances the relative blacking due to a flaw. If the ray passing through an object of thickness t has an intensity I_1 on reaching the film, and a ray which passes through a cavity of diameter x has an intensity I_2, then—

$$I_1 = I_0 e^{-\mu t}$$

$$I_2 = I_0 e^{-\mu(t-x)}$$

Hence $$I_2/I_1 = e^{\mu x}$$

Thus for a given value of x the ratio is greater the greater is μ, and it has already been stated that μ is greater the longer the wavelength, i.e. the lower the high tension. Too low a tension cannot be used, however, since the exposure becomes too long. The determinant for a suitable high tension is mainly the economic one of reasonable exposure time.

Exposure times are reduced by the use of—

1. Double-coated films (since the rays go through the film, they can be made to blacken the front and back).

2. Intensifying screeens. These screens have a coating of fluorescent salts and are pressed into contact with the film, one on the front and one on the back. Under X-rays they fluoresce—mainly ultra-violet light—so adding to the intensity of blackening. In effect, both (1) and (2) are means of trapping additional energy from the beam. Screens may reduce the exposure by fifty times or more, but are not used in every case, as there is a slight loss of definition.

The fact that the energy absorbed from the incident beam partly reappears as scattered radiation is one of the greatest difficulties that have to be encountered in the practice of radiography. Some of this radiation, originating as it does at every point in the object, reaches the film and causes general blackening like a fog over the entire image. Various means have been devised to combat the difficulty, but none is entirely satisfactory for the voltages used in industry. It is possible to absorb the scatter reaching the film from the object by interposing a thin sheet of metal, which acts as a filter. At present the radiographer must select the absorbing power of the filter on empirical grounds in the absence of a precise analysis of all the factors concerned.

THE PHILIPS " MACRO 100 " INDUSTRIAL X-RAY OUTFIT

This is described as being representative of the specialized apparatus developed for non-destructive testing.

The Apparatus. The complete apparatus consists of a metalclad rayproof and shockproof X-ray tube fed from a high-tension transformer

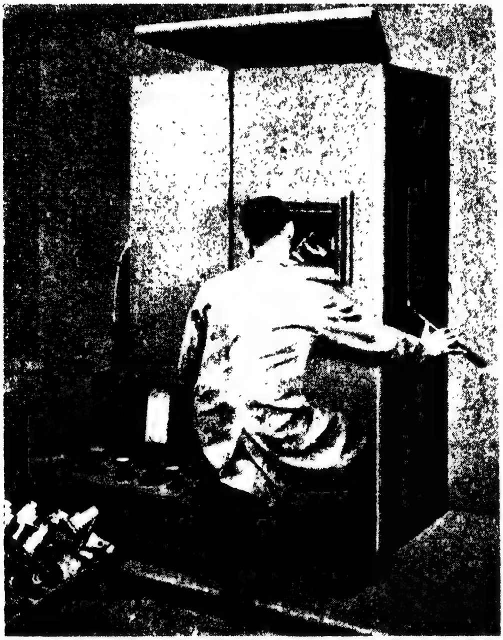


FIG. 85. "MACRO 100" VERTICAL SCREENING STAND FOR VISUAL EXAMINATION

(*Philips Lamps Ltd.*)

by means of shockproof flexible cables. Tube and transformer together form the X-ray generater proper. To permit of maximum working convenience the control system is arranged in the form of a separate unit. This is coupled with the transformer by means of an armoured flexible cable.

The complete outfit therefore consists of—

1. The X-ray tube.
2. The energizing transformer.
3. The control unit.

The mobile tube stand as shown in Fig. 86 is not part of the unit, but is an essential form of mounting in practically every type of application. With this stand the X-ray beam can be adjusted for height and angulated in any direction.

The X-ray Tube. In an X-ray tube, the smaller the source of the radiation, i.e. the focal spot, the sharper the definition in the radiograph

or in the screen, as the case may be. For this reason the focus is made as small as possible consistent with the capacity of the tube for dissipating the heat generated. In the "Macro 100," the optically effective focus is only 0·6 mm. square. The tube window is of low-filtration glass.

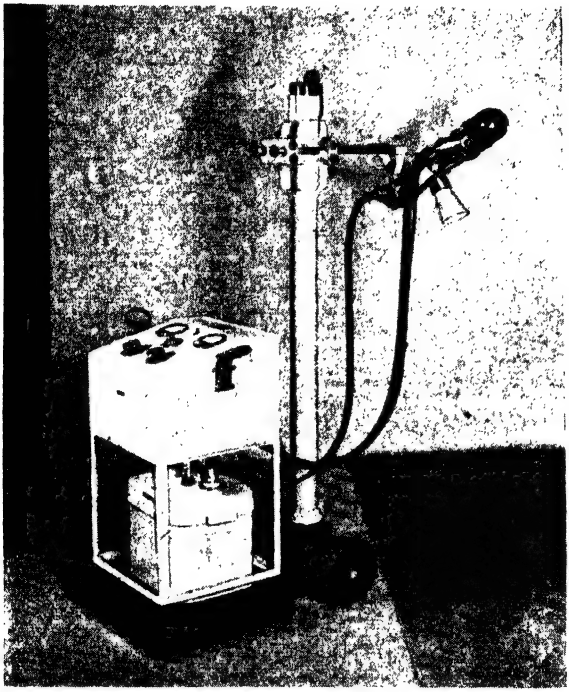


FIG. 86. THE "MACRO 100" UNIT WITH MOBILE STAND
(*Philips Lamps, Ltd.*)

While the apparatus is in use, adequate cooling is provided by means of a forced-draught cooling attachment.

The Energizing Transformer. The high-tension transformer is oil-immersed and contained in a steel tank. Insulated sockets are provided in the top for the reception of the shockproof cables, a mechanical interlock preventing their removal while the current is on. Also the main supply cannot be connected with the apparatus unless the cables leading from the tube are properly coupled up.

The Control Unit. This is specially designed with a view to portability, and is therefore provided with carrying handles. A system of voltage compensation is incorporated whereby the input to the machine can be kept constant irrespective of possible changes in the line voltage. This ensures stable operating conditions and the use of a standardized technique for any particular application.

A penetrating selector is also provided, permitting the voltage to be varied from 30 kV. to 100 kV. in eleven stages.

The X-ray beam can be switched on or off by a single switch at any predetermined penetration.

Scope of the "Macro 100." The following table gives the scope of the unit both for visual and radiographic penetration.

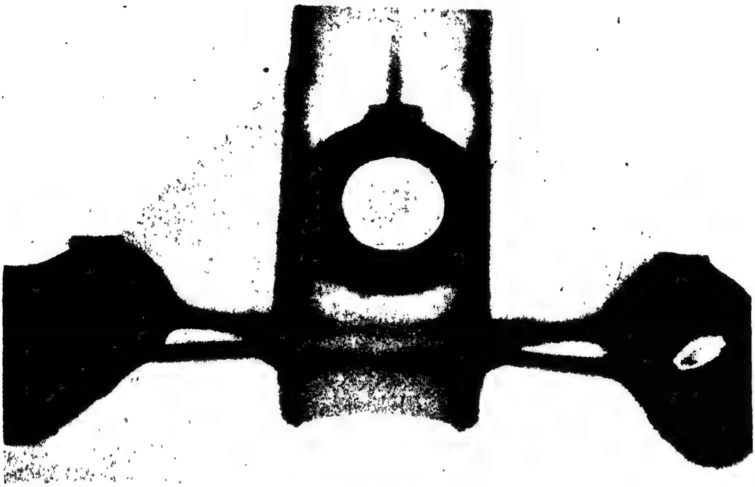


FIG. 87. RADIOGRAPH OF ALUMINIUM ALLOY CASTING, SHOWING POROSITY

TABLE XVI

Material	Radiographic penetration	Visual penetration
Aluminium alloys	4 in.	Up to 1 in.
Magnesium alloys	Up to all practical thicknesses	Up to all practical thicknesses
Steel	$\frac{5}{8}$ in.	$\frac{1}{16}$ in.
Copper	$\frac{3}{8}$ in.	—
Wood	Up to all practical thicknesses	Up to all practical thicknesses
Plastics	Ditto	Ditto

For increased penetrations a 150 kV. and a 300 kV. unit are available.

A Method of Routine Inspection. The "Macro 100" can be arranged for the examination of articles by the visual method as they pass along a conveyor belt. The X-ray tube is mounted under the conveyor, with the beam projecting upwards. Above the articles is a fluorescent screen, from which the X-ray image is reflected into an oblique mirror, thus permitting the operator to examine the objects on the conveyor from a comfortable position. By means of a simple trigger mechanism attached to the

conveyor, faulty articles can be rejected. A diagrammatic arrangement of the routine inspection method is shown in Fig. 88.

CRACK DETECTION

The magnetic method of crack detection in ferrous material has almost entirely superseded the older visual-type inspection with the aid of magnifying glasses. In highly stressed engine components particularly,

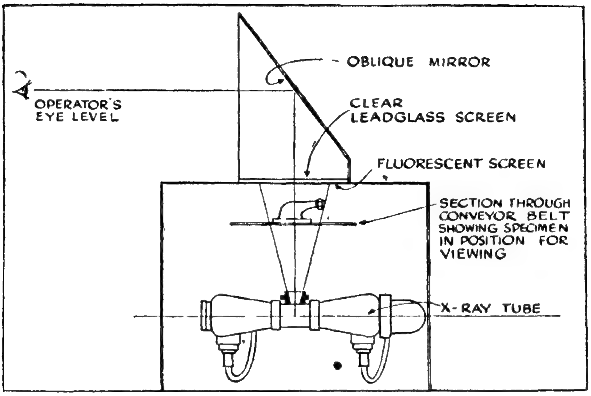


FIG. 88. ARRANGEMENTS FOR VISUAL EXAMINATION OF OBJECTS ON A CONVEYOR BELT

crack detection during manufacturing process and complete overhaul of engines is of paramount importance.

From the manufacturing point of view, the elimination of faulty material effects considerable saving in production time and costs, while from the aircraft operating side, increased serviceability and decreased maintenance will result from the rejection of components which during overhaul are found to possess cracks of a dangerous character. Numerous cracks may exist initially which cannot be detected by ordinary visual viewing, but the magnetic method will show them clearly.

Assuming initially sound material, cracks may originate in components due to (1) incorrect processing, e.g. rash grinding, drilling, or machining, especially when dealing with intensely air-hardening steels, (2) faulty heat treatment, giving rise to hardening cracks, and (3) fatigue.

The action of fatigue is often initiated by small cracks due to cases under (1) and (2), and provided the design of a stressed component is correct and that cracks due to the above-mentioned causes are not present, there is little likelihood of fatigue failure.

From the manufacturing point of view the cracks will only occur due to incorrect processing or heat treatment, as the fatigue type of crack is only apparent after the particular component has been used in service.

Detection of the crack in a partly finished component will give a clue to its origin and thus prevent repetition of cracks due to the same

cause. Detection of fatigue cracks during engine overhaul will enable the design of the part to be modified if that is the contributory factor.

All three types of crack can be detected by simple etching, but obviously the use of even weak acid solutions is objectionable, especially when dealing with finished parts.

The etching process is more applicable for detection of heterogeneity.

The Johnson-Fel Crack Detector. This instrument, the standard type of which is illustrated in Fig. 89, is a magnetic crack detector used for the testing of ferrous material with the exception of the non-magnetic

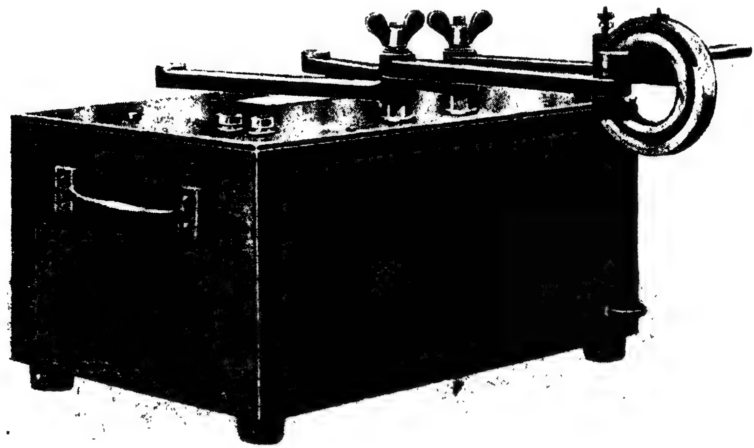


FIG. 89. THE JOHNSON-FEL STANDARD-TYPE CRACK DETECTOR

austenitic type of steel. Magnetizing of the specimen can be carried out in two ways—

1. By the passing through it of an alternating current,
2. By means of an external magnetic field.

In both cases the current flow is only momentary, so that crack detection is by means of residual magnetism. Method 2 is used for components having holes by means of which they can be threaded on to a special jig bar.

The Operation of the Machine. On connecting up to the mains, a pilot lamp will light. The component to be tested, if not suitable for threading on to the jig bar, is laid across the adjustable contact arms, and the circuit-closing push-button switch depressed and quickly released. If the component has been correctly magnetized, a second lamp will light.

On removing the magnetized part from the contact arms it is covered with special detector ink containing iron dust, and any cracks will be clearly indicated by the dust adhering to them.

It should be noted that only cracks breaking out on the surface can be detected by this means.

Procedure for Long Bars, etc. When these are longer than can be accommodated across the contact arms, it is necessary to have a pair of cables, $\frac{3}{8}$ sq. in. to $\frac{1}{2}$ sq. in. cross-section, attached to the posts provided on top of the machine. The cables must be attached to suitable copper

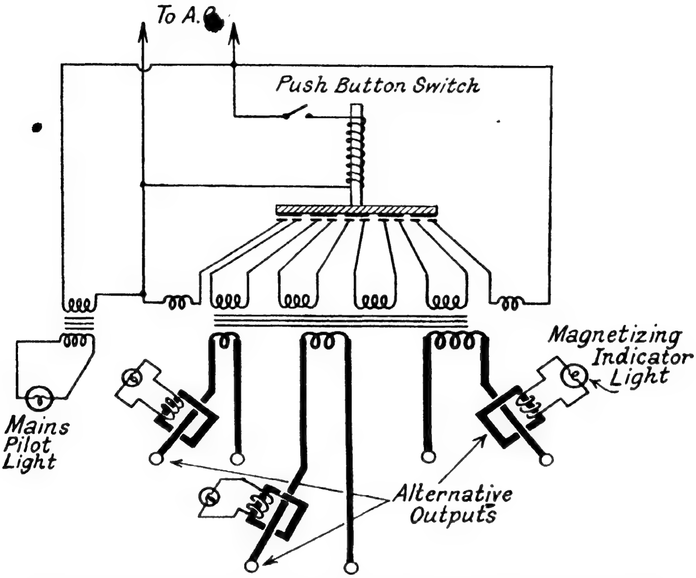


FIG. 90. CIRCUIT DIAGRAM OF JOHNSON-FEL TYPE R MACHINE

clamps to make contact with the ends of the bar. Bars of great length can be examined section by section, by moving the clamps along.

Crankshafts can be tested by supporting them in vee blocks lined with copper or copper gauze, and connecting the cables to the blocks. It is, of course, necessary to insulate the vee blocks if they are standing on a marking-off table. End-faces of large-diameter bars can be examined by making contacts diametrically opposite.

Rings, Ball Races, Gears, Gudgeon Pins, etc. These are threaded on the movable jig bar, the bar placed back in position, and the switch pressed, when the part is ready for examination by the detector ink. The Johnson-Fel ring jig is one method by which these components can be examined for defects in all directions and in one operation.

Castings and Welded Structures. Suspected portions of any large casting may be examined by applying a copper contact at each end.

Examination of fillet welds can be satisfactorily carried out by means of copper contacts shaped to fit close into the corner of the fillet and being placed not more than 18 in. apart.

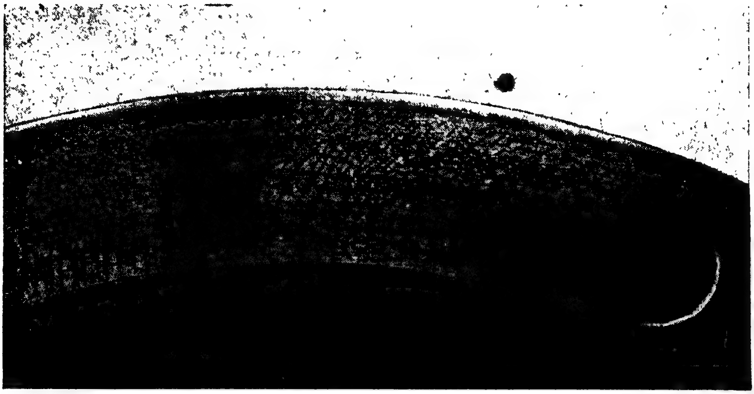


FIG. 91. CRAZY CRACKS ON A GROUND RING

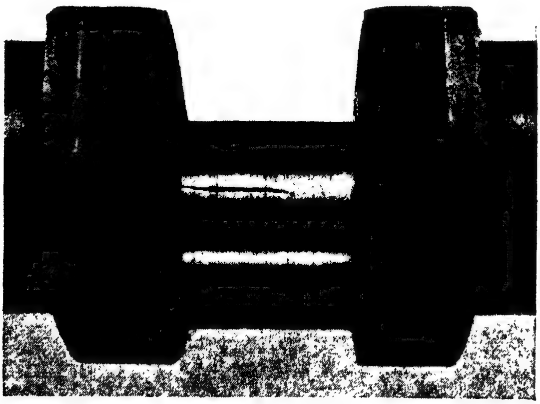


FIG. 92. CRANKSHAFT WITH CRACKS ON THE PIN

The "Universal Four" Machine. To obviate the necessity for providing a variety of jigs and clamps to grip the component under test and to ensure a good area for electrical contact in addition to the capability of handling a great variety of work on a self-contained unit, the "Universal" type machine illustrated in Fig. 93 has been designed.

The special features of this unit are the sliding heads fitted with resili-

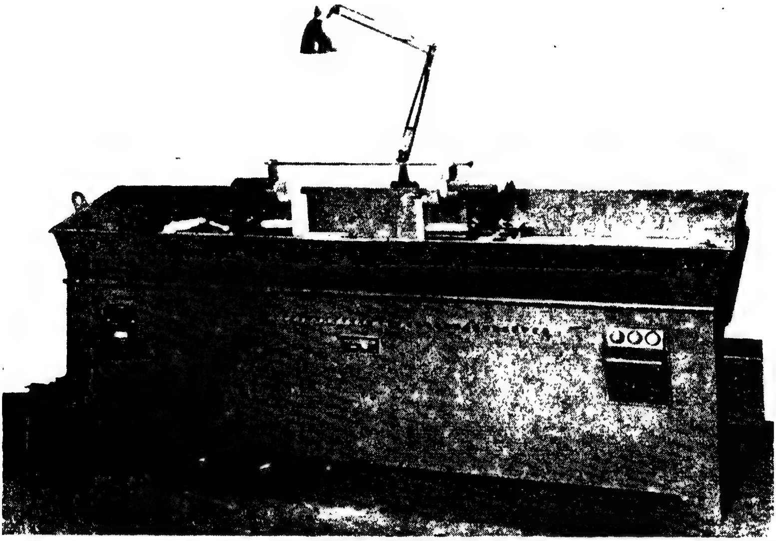


FIG. 93. UNIVERSAL FOUR MACHINE SHOWING RING JIG BAR IN POSITION, ALSO RESILIENT CONTACT HEADS AND VEE BLOCK POLE PIECES
(*Fel. Electric Ltd.*)

ent contact pads, the heads being adjustable within a range of from 2 in. to 4 ft., and the self-contained tank for the detecting fluid.

When any component is gripped tightly between the resilient pad faces the pressure causes it to sink into the face of the pad, and the copper gauze contact faces follow the contour of the component, thereby substantially increasing the area available for good electrical contact.

The ring jig, which is formed by a copper rod, is placed in the slotted portion of the sliding head at a point above and adjacent to the resilient contact pads. When not in use the rod is placed in clips provided on the side of the machine.

In use, the operator selects the test voltage by means of a rotary switch, depresses and then sharply releases a switch button, when an automatic switch completes the electrical circuit which applies the magnetizing current for the correct period of time. A green lamp which glows during the period of magnetization is a visual check that the electrical circuit formed by the contact faces and the specimen under test is complete.

The Residual Magnetism. The composition of the material determines

the length of time during which the magnetizing effect will last. On a very low-carbon steel it will remain for about six hours, whereas in the case of higher-carbon steels and alloy steels it will remain for three months or more.

Aperture type de-magnetizers are available, and these can be built with a runway and trolly which facilitates the handling of work. A small platen type de-magnetizer is also made for de-magnetizing parts of a structure when they are in their assembled position.

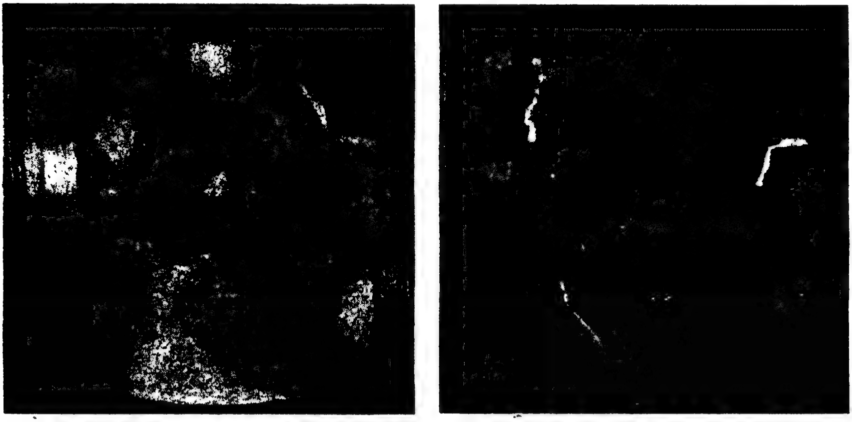


FIG. 94. PHOTOGRAPHS OF LIGHT ALLOY PUMP HOUSING
(*Collodial Research Laboratories Ltd.*)

It should be remembered that specimens magnetized on a Johnson-Fel crack detector will only show a field across a crack, and in the absence of cracks will appear non-magnetic when tested with iron filings. Components such a splined shafts, gears, etc., will indicate a magnetic field between the splines and from tooth to tooth of the gear.

Types of the Johnson-Fel Detector. Several types other than those illustrated are manufactured for various applications. In particular a special Bar Machine has been developed for testing bars between $\frac{1}{2}$ in. and 3 in. diameter, and from 4 ft. 6 in. to 15 ft. in length. On this model an electrically-driven pump is incorporated which, in addition to keeping the particles in the detecting fluid in a state of suspension, also provides a flow of fluid to a hand spray gun with which the operator floods the bar after it has been magnetized.

The Glo-Crack System. This system of crack detection is unique in as much as it is applicable to a wide range of materials such as ferrous and non-ferrous alloys, plastics, and ceramics.

Hitherto, apart from X-ray examination, there has been no really satisfactory means of crack detection in non-ferrous alloys comparable with the magnetic system employed for the magnetic ferrous materials.

The Glo-Crack system is based upon the phenomenon that whereas a non-fluorescent material illuminated with ultra-violet light will appear

black or purple, a crack or flaw filled with a fluorescent filler, will glow vividly. Inspection is thus facilitated as the defect will be the most outstanding feature.

The actual method consists in dipping the parts to be examined in a fluorescent liquid of a particularly penetrating nature contained in a thermostatically controlled bath. After immersion the parts are drained and transferred to an inhibiting bath for a few moments, after which they are removed and allowed to dry. They are then passed under the ultra-violet light and turned about so that all surface areas are exposed to the activating rays. Where cracks exist they will be disclosed by the characteristic glow of the fluorescent material which has been trapped at the edges of the cracks. Both the solutions are inert, and have no deleterious effects on the materials tested which are efficiently degreased by immersion in the fluorescent solution.

Equipment. The equipment consists of two tanks separated by a draining rack. The first, containing the fluorescent liquid, is provided with a thermostatically controlled heating element which is set to maintain a temperature of about 70° C. in series with a green pilot lamp. The second tank contains the inhibiting liquid the purpose of which is to remove surplus fluoresecnce.

The ultra-violet light is provided by an 80 Watt mercury vapour discharge lamp, and a black glass filter.

Three standard models are available. Model No. 1 is primarily a laboratory unit, but is sufficiently robust for use where the quantity of materials to be tested is small. The two baths each take about three gallons of solution. Model No. 2 is capable of dealing continuously with and testing pieces up to 23 in. × 35 in. × 12 in., and Model No. 3 up to 58 in. × 35 in. × 18 in.

For the inspection of the interior of hollow components a U-tube source of ultra-violet light can be provided.

Fig. 94 shows cracks revealed in a light alloy pump housing when using the Glo-Crack System.

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